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ABSTRACT

Designed for student use in "System Sizing, Design, and Retrofit," one of 11 courses in a 2-year associate degree program in solar technology, this manual provides readings, exercises, worksheets, bibliographies, and illustrations for 13 course modules. The manual, which corresponds to an instructor guide for the same course, covers the following topics: (1) design considerations and parameters; (2) load calculation factors and procedures; (3) thermal load analysis--space heating and space cooling; (4) thermal load analysis--service water; (5) sizing and selection of the collector array--manual method; (6) sizing and selection of the storage system; (7) sizing and selection of subsystem components; (8) system controls and protective devices; (9) equipment and component specifications and selection; (10) retrofit considerations; (11) programmed system sizing--analysis and design; (12) swimming pools, spas, and hot tubs; and (13) installation, maintenance, and operational considerations. (AYC)

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SOLAR ENERGY

SYSTEM SIZING, DESIGN, AND RETROFIT

Student Material

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NAVARRO COLLEGE
CORSICANA, TEXAS

JC 830 236

Student Material

SOLAR ENERGY

SYSTEM SIZING, DESIGN, AND RETROFIT

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FIRST EDITION

1982

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SYSTEM SIZING, DESIGN, AND RETROFIT
Student Material

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PREFACE

The United States is facing one of its most challenging decades in recent history. Fuel supply and inflationary prices have forced us to consider alternate energy sources as a means of preserving our standard of living, industrial society, and economic stability. One such alternative is solar.

Presently, foreign crude oil provides the raw material for about one-half the liquid fuel production in the U.S. Political instability in foreign oil-producing countries underscores the need to decrease our ever-growing dependency on foreign energy sources and to lessen our vulnerability to such imports. Solar energy as an alternate can be used as a renewable domestic energy source and to supplement our increasing appetite for oil.

To help bring about the potential for solar energy, there must be a cadre of trained technicians to design, install, troubleshoot, and market solar energy so that the consumer can feel comfortable in the market's ability to service and react to his/her solar energy needs.

With the support of the National Science Foundation, Navarro College, in consortium with North Lake College, Brevard Community College, Cerro Coso Community College, and Malaspina College, has developed and pilot tested a two-year associate degree curriculum to train solar technicians. It can be duplicated or replicated by other educational institutions for their training needs.

The two-year technician program prepares a person to:

1) apply knowledge to science and mathematics extensively and render direct technical assistance to scientists and engineers engaged in solar energy research and experimentation;
2) design, plan, supervise, and assist in installation of both simple and complex solar systems and solar control devices;
3) supervise, or execute, the operation, maintenance and repair of simple and complex solar systems and solar control systems;
4) design, plan, and estimate costs as a field representative or salesperson for a manufacturer or distributor of solar equipment;
5) prepare or interpret drawings and sketches and write specifications or procedures for work related to solar systems; and
6) work with and communicate with both the public and other employees regarding the entire field of solar energy.

This curriculum consists of nine volumes:

1) an Instructor's Guide for the eleven solar courses, to include references, educational objectives, transparency masters, pre-tests and post-tests, and representative student labs;
2) an Implementation Guide addressing equipment, commitment, and elements to be considered before setting up a solar program;
3) Student Material for each of seven of the core solar courses:
    a) Materials, Materials Handling, and Fabrication Processes;
    b) Sizing, Design, and Retrofit;
    c) Collectors and Energy Storage;
    d) Non-Residential Applications;
    e) Energy Conservation and Passive Design;
    f) Codes, Legalities, Consumerism, and Economics;
    g) Operational Diagnosis.

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Appreciation is extended to thos members who served on the Advisory Committee and provided valuable input to module content, text format, and the pilot test plan and evaluation:

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My personal thanks go to Bill Bolin for all his help, and to my wife and children for their emotional support.

USE OF THE STUDENT MATERIALS

The intent of this manual is for student use as a supplement to the instructor's guide for the same course. It contains readings, exercises, worksheets, bibliographies, and illustratons to reinforce the concepts contained within this particular course of study. Each student materials manual is written in a similar format but differs in some details due to the nature of the course and the subject matter covered.

Pretests, posttests, and lab exercise are not contained in this manual. Refer to the instructor's guide for this course to find these items.

Student materials manuals are supplied for seven of the eleven solar courses in this project. The four not included are: Introduction to Solar Energy, Energy Science I, Energy Science II, and the Practicum.

The pagination code is used as follows:

I -- the Roman numeral coordinates with the Roman numeral of the instructor's guide.

S -- the "S" signifies that the page is from the Student Material.

5 -- the Arabic number reflects the specific page within this manual, numbered sequentially throughout.

SIZING, DESIGN AND RETROFIT

DESIGN CONSIDERATIONS AND PARAMETERS

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

DESIGN CONSIDERATIONS AND PARAMETERS

Before the design process begins, one must look at the limitations placed upon the design by existing building and environmental conditions. Examples of these conditions are as follows:

A. What energy conservations methods should be implemented before solarization of the structure; i. e., insulation, weatherstripping and caulking, thermal efficient glazing?

B. Awnings, shades, trees, shrubbery, and other exterior treatment of the structure that could improve the thermal efficiency of the building. Methods for achieving optimum thermal efficiency are presented in depth in the course Energy Conservation and Passive Design Concepts.

Also, one should make note of the building site and orientation since these factors will affect the choices that will be made in designing the solar systems. The north/south exposure and east/west exposure are important factors in the limits placed upon the design. Shading from trees, buildings and other surrounding structures are critical in the performance of the solar system.

One means of determining potential shading problems is through the use of various solar siting instruments that allow one to determine the "solar window" for a particular structure. The figure below illustrates how such a device would be useful in determining the days and times that trees, buildings, earth strata, etc., might block out the rays from the sun. These structural shading problems change with the seasons (see figs. 1-1 through 1-4 on following page).

Other factors that affect design decisions are roof pitch, roof type and roof mounting space available at an acceptable orientation to the sun. Assuming the design is to be a roof-mounted collector array, the roof structure (pitch, angle and type) must be examined to determine whether or not additional roof load from the collectors will require increasing the load capacity of the roof. Also, low pitched roof may require a collector rack with possible furring for esthetic reasons. Collector tilt angle is far more critical to optimum performance than orientation angle.

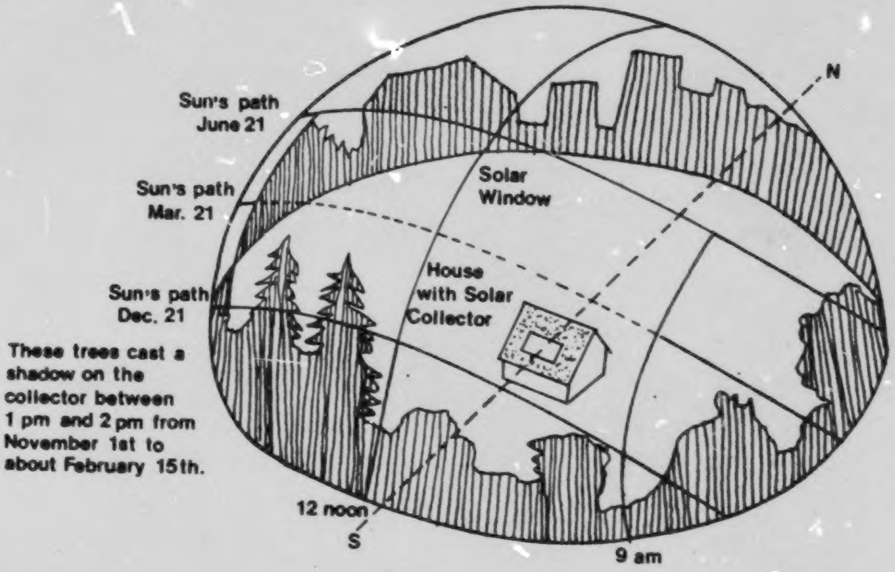

Sun's path June 21
Sun's path Mar. 21
Sun's path Dec. 21
These trees cast a shadow on the collector between 1 pm and 2 pm from November 1st to about February 15th.
Solar Window
House with Solar Collector
N
12 noon
S
9 am

Figure 1-1: The Solar Window

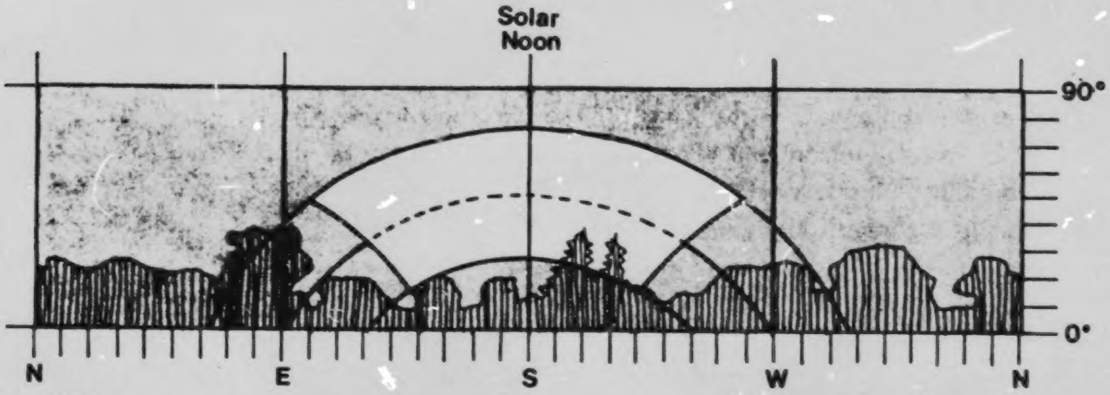
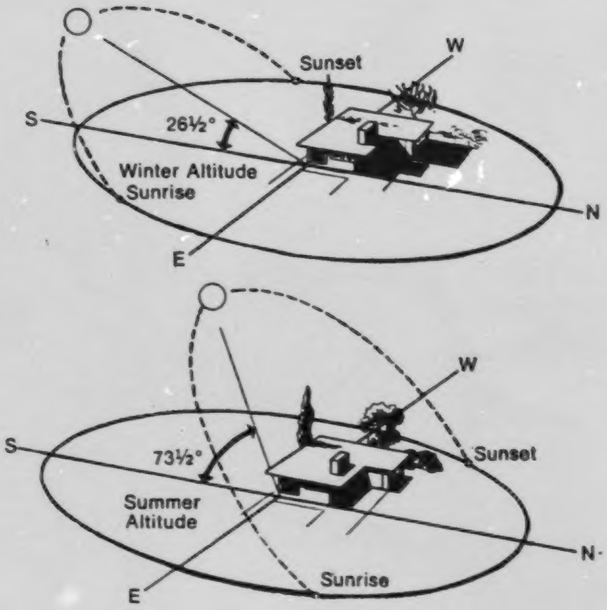

Solar Noon
90°
0°
N
E
S
W
N

Figure 1-2: Mercator Projection


Sunset
W
S
26½°
Winter Altitude
Sunrise
N
E
W
S
73½°
Sunset
Summer Altitude
N
Sunrise
E

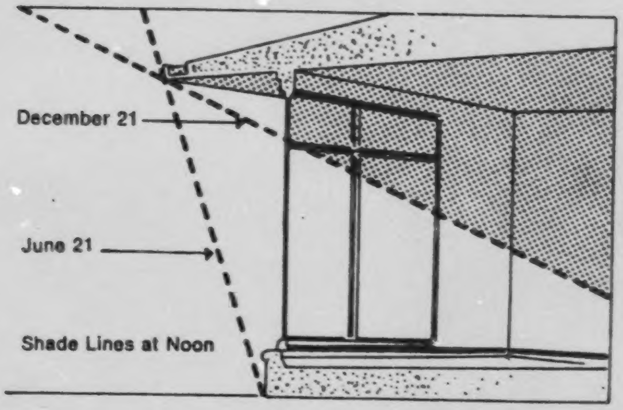

December 21
June 21
Shade Lines at Noon

Fig. 1-3: SUN'S position in sky affects amount of heat absorbed. Winter sun is hotter but lower in the sky than summer sun. Angles shown are for 40 deg. N Latitude (Chicago).

Fig. 1-4: High summer sun is blocked out by roof overhang while in winter, lower sun permits rays to penetrate house.

Another item for consideration is the hourly position of the sun, which is clearly illustrated in Figure 1-5. Summer days have more hours of sunlight and the area exposed to this solar radiation is greater.

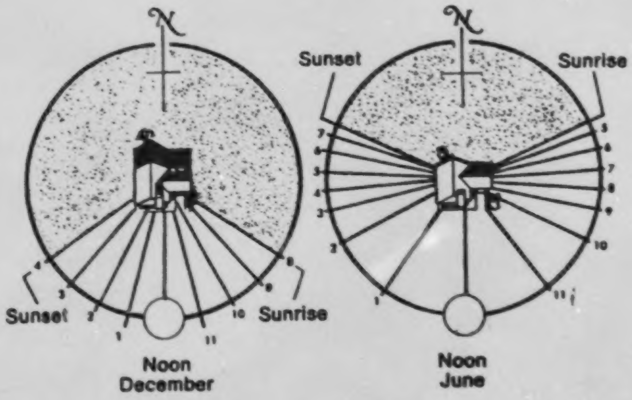

N
Sunset
Sunrise
Noon
December
N
Sunset
Sunrise
Noon
June

Fig. 1-5: With the sun higher in the sky in summer there are more daylight hours in summer than in winter. Note time of winter and summer sunrise and sunset.

Early morning sun is at a very low angle and the solar rays must pass through a larger thickness of atmosphere than at noon time (see Figure 1-6). This is a prime reason why noon sun is "stronger." If the sun was tracked with a pyrheliometer through the day, the Btu's received would vary in a manner shown in Figure 1-7.

If the solar energy received each hour was plotted on a fixed horizontal surface that same day, the pattern would be indicated by the solid line in Figure 1-8.

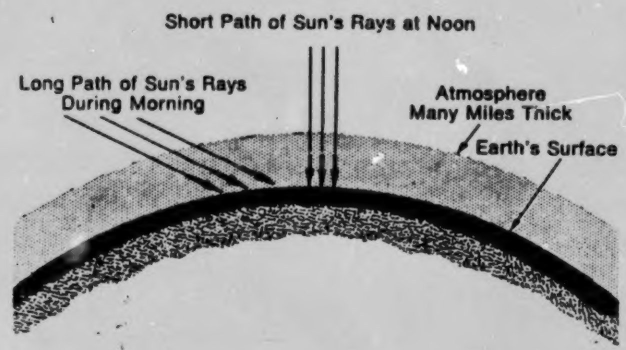

Short Path of Sun's Rays at Noon
Long Path of Sun's Rays During Morning
Atmosphere Many Miles Thick
Earth's Surface

Fig. 1-6: Interception of sun's rays by earth's atmosphere.

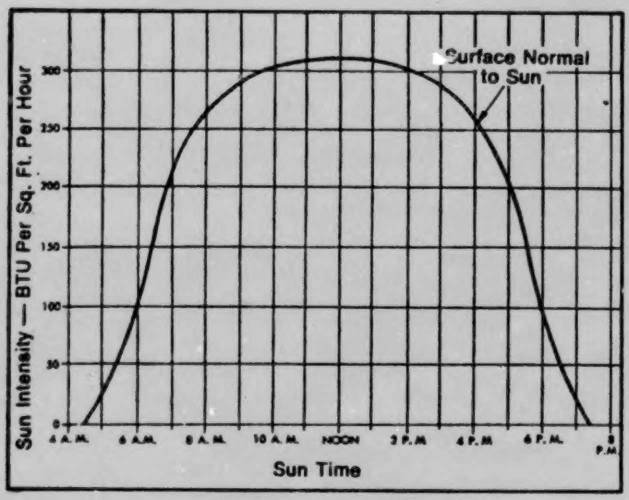


Fig. 1-7: Energy is received by a surface kept normal to sun in summer

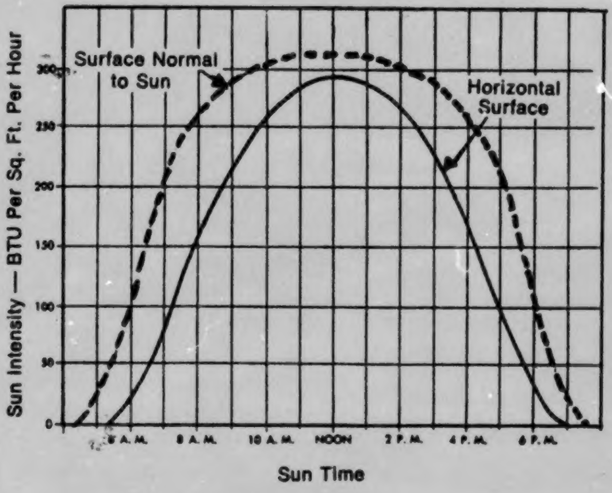


Fig. 1-8: Energy received by horizontal surface is less than that received by a surface kept normal to a summer sun.

As expected, the greatest intensity still occurs during the noon hour, but the energy received by the horizontal surface was less than in the previous case when the sun was tracked (dotted line is from Figure 1-8). The curve indicates the readings were taken on a clear day since it is smooth. The presence of clouds would have caused breaks in the curve.

While roofs are generally considered to be horizontal or tilted surfaces, building walls are not. However, walls are able to absorb solar radiation just like horizontal surfaces. The only difference is that their exposure time to the sun's rays are different. (See Figure 1-9). Another item of interest in figure 1-9 is that the north wall is not included. But, since the north walls of buildings in the northern hemisphere are not exposed to direct solar radiation in winter, a reading would prove useless since it would be a flat line zero. North walls can receive some diffuse radiation, however; thus, the surface direction (north, south, east, west) and the surface tilt (horizontal, vertical, etc.) all affect the amount of solar radiation actually intercepted.

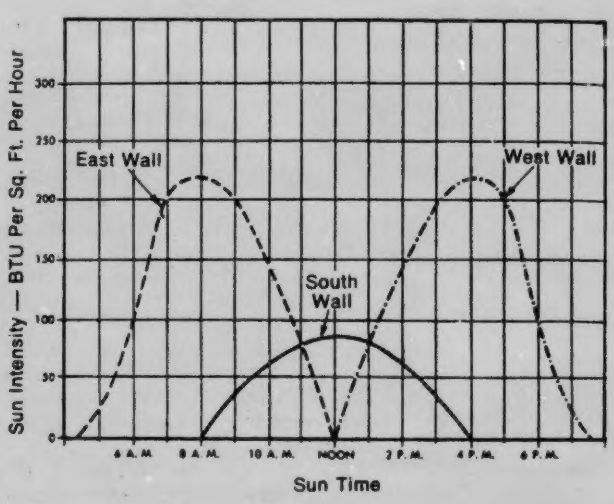


Fig. 1-9: Solar radiation falling on a building's walls in summer.
Note times of maxumum intensity.

As mentioned previously, the amount of solar radiation incident upon any surface (horizontal, vertical or otherwise) is affected by the orientation to Solar South and shading patterns. It is necessary to predict the effects of shading at various seasons and hours of the day. The Solar Site Selector enables one to make these predictions.

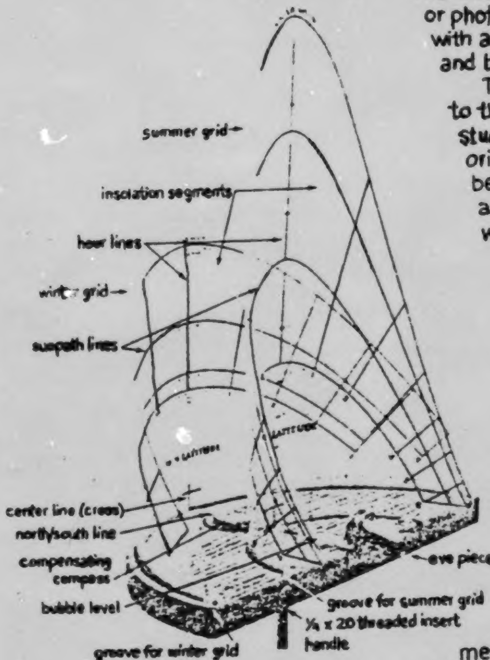
Solar Site Analysis with the Solar Site Selector
(in the Northern Hemisphere)

The SOLAR SITE SELECTOR calculates solar access and demonstrates shading patterns for any given site or surface throughout the year. This is done by using silk-screened grids that function as solar windows.

When viewed through the 180° eyepiece, the sunpaths, hour lines and insolation segments are superimposed on the site being studied instantly depicting shading patterns and making possible a simple visual calculation of hours of solar access. The instrument also enables a quick computation of the percent of occlusion and percent of incident solar radiation (insolation) for that site.

The instrument fits a handle (included) or photographic tripods and is oriented with a built-in compensating compass and bubble level.

Take the SOLAR SITE SELECTOR to the site or building you wish to study for solar access, siting and orientation. Accurate readings can be made for the entire year at any time of day, in clear or cloudy weather.



Solar Site Selector

Set-Up

Step 1. Screw baseplate into handle or tripod.

Step 2. Select the proper grid or grids. Whether to use the winter or summer grid, or both, is clearly determined in most cases by the type of solar measurements needed. If you are unsure, however this booklet contains further discussion about the proper use of winter and summer grids in later sections. Once determined, choose the grid or grids corresponding to your nearest latitude.

Step 3. Insert grid firmly in the groove by pushing down so grid touches bottom, then press the front edges back so the grid rests against the back rim of the groove. If inserted properly, the instrument can be held upside down without the grid moving. Grids can be trimmed on curved outer line for easier use.

Step 4. Align center line (cross) on grid with north/south line on wood baseplate.

V-S-9

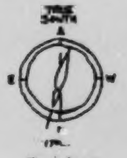
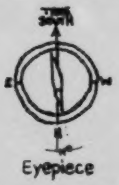
Orientation

•Since solar calculations are based on true south, compensate for the difference between true north/south and magnetic north/south by:

Step 1. See Isogonic Chart and determine your magnetic declination Users outside continental US will need to consult local sources

Step 2. Adjust compass by rotating the clear plastic center until the black arrow points to degree of declination from North. (If you are in the eastern portion of the US you have a western declination. See map Subtract your angle of declination from 360 to determine the proper compass settir).

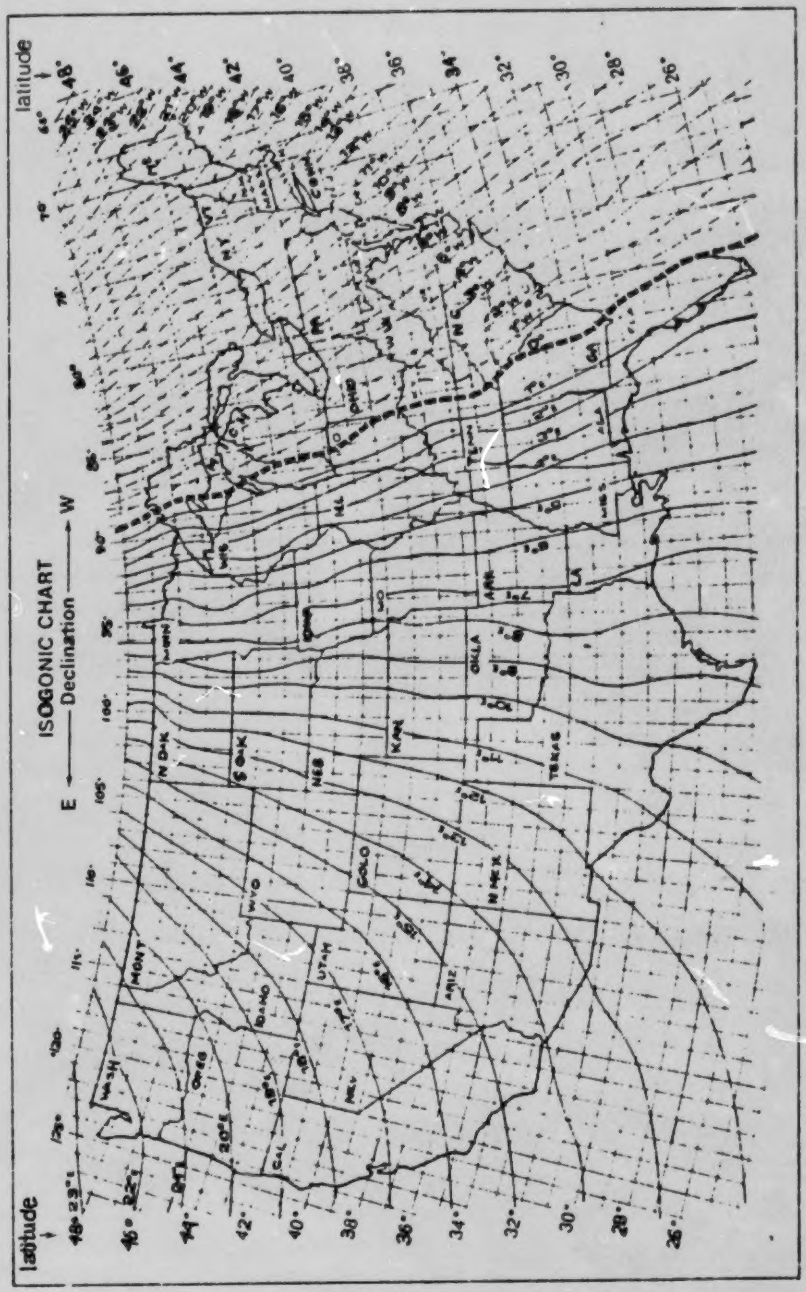
Step 3. Now align the red magnetic needle (N) with the black arrow pointing to the degree of declination. When looking down on the SOLAR SITE SELECTOR while facing south, the properly aligned arrows will be pointing in the general direction of the eyepiece.

 Eyepiece Example: Compass reading for San Francisco, 17° east declination	 Eyepiece Example: Compass reading for New York, N.Y.. 11° west declination. Set arrow at 349° (360°-11°)
For eastern declinations the needle (N) will be left of center.	For western declinations the needle (N) will be right of cer

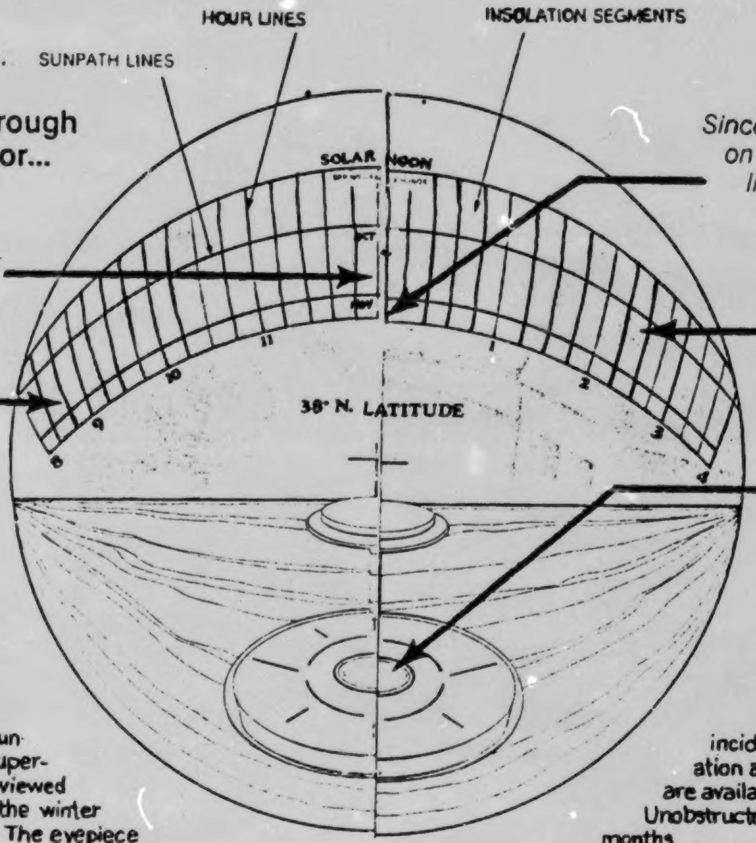
The north/south line on the paseplate is now aligned with true south.

Step 4. For tripod use, level (using bubble level) and secure position For hand held use leveling is done while viewing through the eyepiece. Without changing compass orientation, raise the instrument, look through the eyepiece, and make sure the bubble in the level is centered within the circles. You are now ready to take a sighting and make an analysis of your site. Be sure the instrument is level and aligned with true south throughout the process.

Note: The compass may be deflected if the instrument is set up under heavy electric wires or near a large mass of metal with magnetic properties.



A simulated view through the Solar Site Selector...



Maximum solar radiation occurs at solar noon. a point instantly determined with the Solar Site Selector.

This deciduous tree provides desirable shade in the summer yet allows sunlight through in the winter when its leaves are gone.

Since the sun's path is lowest on Dec. 21, objects below this line are not obstacles to solar access.

The shading problem caused by this structure can be minimized by reorienting the active or passive system slightly to the east.

This bubble level can be seen through the viewfinder to easily insure accurate siting analysis.

Sighting

Looking through the eyepiece, notice the sunpath lines, hour lines and insolation segments superimposed on the site. Also notice any objects viewed through the grid or grids that extend above the winter solstice line and into the insolation segments. The eyepiece swivels from side to side for easy viewing of early morning and late afternoon hours.

Example of Determining Solar Access and Shading Patterns

The illustration on this page simulates the type of shading objects that may be encountered as viewed through the SOLAR SITE SELECTOR All objects that extend above the winter solstice line will cast a shadow on the site at the time of day indicated by the hour lines.

As seen through the winter and the deciduous trees (shown here as viewed in the summer with full foliage) will have shed their leaves and be casting a partial "twiggy" shadow from 8 a.m to 11 a.m throughout the winter months. The house will be casting a permanent solid shadow on the site from approximately 1:30 p.m. through the rest of the afternoon during winter months. There are no obstructions going beyond the spring-fall equinox line into the summer months. However, in other applications sighting with the summer grid may be desirable.

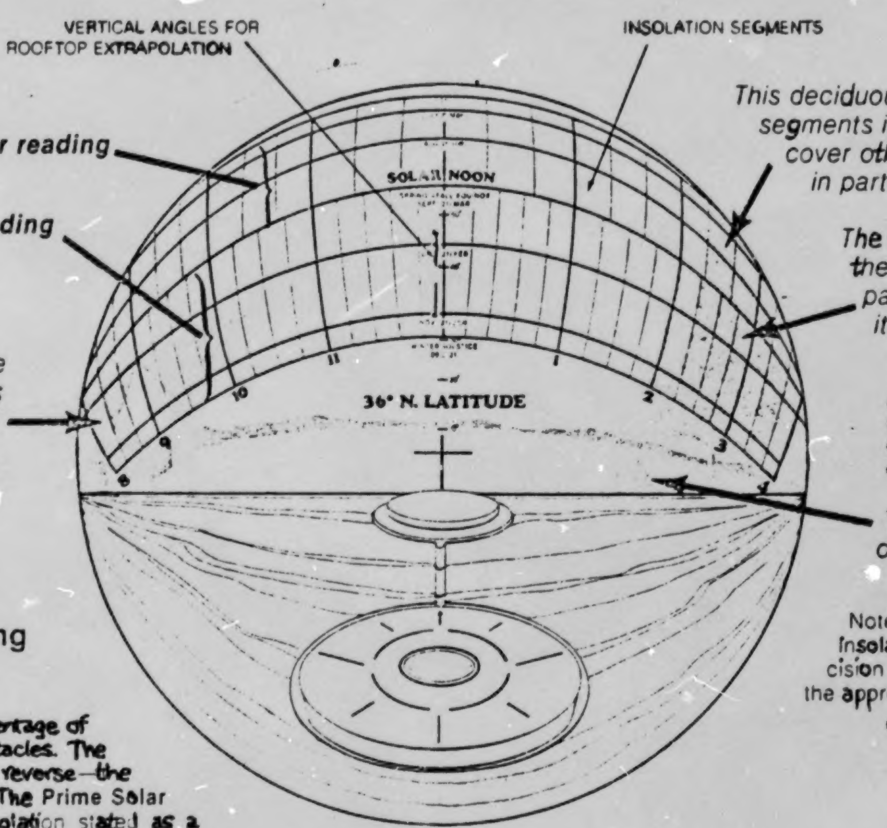
In this illustration there are 2 1/3rd hours of incident solar radiation plus 3 hours of partial radiation available at the winter solstice Over 5 hours are available during both the spring and fall equinoxes Unobstructed radiation is available during the summer months

Without the elimination of at least the largest deciduous tree this site has minimal solar access. To be most efficient and cost effective solar glazing should have 5 to 6 hours per day of unobstructed sunshine at the winter solstice. Since much more incident solar radiation is absorbed between the hours of 10 a.m. and 2 p.m. (as compared to other hours). It is crucial to avoid shading patterns in these four mid-day hours.

Possible future obstructions to solar access should also be considered such as immature trees not yet reaching the winter solstice line or possible future construction to the south

Read "Facts to Consider" and complete the assessment of your site or building.

Note: Obstructions that cause shading problems come from the ground level up. For most purposes, a reading on the winter solstice line with the winter grid is sufficient to determine optimum solar access and orientation. Where cooling criteria are as important as heating in design, construction or installation, use of both a winter and summer grid is recommended. For pool systems concerned only with summer heating a summer grid is advised.



Summer grid for summer reading

Winter grid for winter reading

This house solidly shades some segments and partially covers others on both the winter grid (for winter PSF) and summer grid (for summer PSF).

This deciduous tree will solidly cover its segments in the summer (and may partially cover others where the foliage extends in parts of other segments).

The tree is in a "twiggy" condition in the winter and thus is counted as partial coverage for the segments it covers or extends into.

***Note.** Deciduous trees generally should be considered as partial shading for winter calculations and solid for summer calculations*

Young Bush—possible future obstruction

Sighting and Calculating for Audits

Percent of Occlusion is the percentage of total sunlight blocked by shading obstacles. The Percent of Insolation is simply the reverse—the percent of total sunlight unobstructed. The Prime Solar Fraction (PSF) is the Percent of Insolation stated as a fraction.

The first step in calculating Percent of Occlusion with the SOLAR SITE SELECTOR involves judging whether insolation segments are solidly covered (70% to 100% shaded) or partially covered (less than 70% shaded). A segment can be partially covered in two different ways: either a solid object covers only a part of the segment, or a non-solid object (e.g. leafless tree, lattice work) covers most or all of the segment. Now add all solidly covered segments to *one-half* the total of partially-covered segments. This total is the Percent of Occlusion.

Next, subtract this total from 100% to obtain the Percent of Insolation. Then divide by 100 to get the Prime Solar Fraction (PSF).

When dealing exclusively with winter measures the PSF can be derived from winter grid calculations alone. When a separate Prime Solar Fraction for summer is needed, use the same procedure with the summer grid. Annual Prime Solar Fractions may be estimated from the winter grid, but it is more accurate to calculate a winter and summer Prime Solar Fraction and average the two. Read the Example of Audit Calculations section.

Note. Not all winter grids have exactly 100 insolation segments. Thus, for greater precision multiply the percent of occlusion by the appropriate factor.

Grid latitude:	Multiply by:
20° - 22°	88
24° - 32°	91
34° - 42°	94
44° - 56°	1.00
58° - 60°	1.03

Summer grids have 100 segments, hence no adjustment is necessary.

Example of Audit Calculations

This illustration simulates a typical scene as viewed through the SOLAR SITE SELECTOR during an audit. The eyepiece must be swiveled to both sides to complete the calculations described below.

To calculate the Prime Solar Fraction (PSF) for winter measures, count all winter grid covered insolation segments as follows. The house solidly covers 10 segments, and partially covers another 5 segments. (The eyepiece must be swivelled to both sides to complete the count.) The tree, without its leaves, partially covers 20 segments. Then the total count is 10 plus (5 – 2) plus (20 – 2), or 23. Thus the Percent of Occlusion is 23%, the Percent of Insolation is 77%, and the PSF is 77. If greater precision is desired, multiply the percent of occlusion by the appropriate figure from the chart above; i.e. for 36°, the adjusted Percent of Occlusion would be 22% (23 × .94) resulting in 78% Insolation or a PSF of 78.

To calculate the Prime Solar Fraction (PSF) for summer measures, count the summer grid segments where obstructions appear. The deciduous tree on the right will have full foliage during the summer months so must be counted as a solid shading object filling at least 6 segments. The roofing of the house protrudes into 4 segments, but only partially, yielding a count of 2. The total of insolation segments segments blocked by shading objects is 8 or 8% occlusion resulting in 92% insolation available during the summer months or a PSF of .92.

Calculating the annual PSF from both the winter PSF and summer PSF is a simple matter of averaging the winter PSF and the summer PSF i.e., .78 + .92 ÷ 2 = .85. When working only with the winter grid a summer PSF may be estimated to arrive at an average annual PSF.

SAMPLE AUDIT WORKSHEET

WINTER CALCULATIONS:		
1. Solidly shaded segments		10
2. Partially shaded segments	=	13
3. Total Unadjusted Percent of Occlusion		23
4. Adjusted Percent of Occlusion	23 x .94 =	22
5. Percent of Insolation	100% - 22 =	78
6. Prime Solar Fraction (PSF)	75 ÷ 100 =	.78
SUMMER CALCULATIONS:		
7. Solidly shaded segments	=	6
8. Partially shaded segments	4 ÷ 2 =	2
9. Total Percent of Occlusion (No adjustment needed for summer grids)	=	8
10. Percent of Insolation	100% - 8 =	92
11. Prime Solar Fraction (PSF)	92 ÷ 100 =	.92
ANNUAL CALCULATIONS:		
12. Winter PSF .78 plus Summer PSF .92	=	1.70
13. Annual Prime Solar Fraction	÷ =	.85

TRACING PROCEDURE

Before making audit calculations, it is highly recommended that a tracing of the shading profile be made. A permanent site record might be desirable for other reasons as well. To do this, trace the shading profile with a non-delible felt tip pen on the back of the winter grid and/or the front of the summer grid (on the opposite side of the silk-screening in each case).

After tracing the shading profile, label obstructions as either solid or partial. (It is also advisable to label plants and trees as either "young" or "mature" for future reference.) The tracing may then be photocopied on a book copier (not sheet fed). The shading profile copy may also be made on traceable paper.

When more than one tracing will be made before a grid can be photocopied or transferred to drafting paper, place a clear reusable mylar sheet against either or both grids before each tracing. Non-indelible felt tip pen ink is easily removed from the mylar for future sightings.

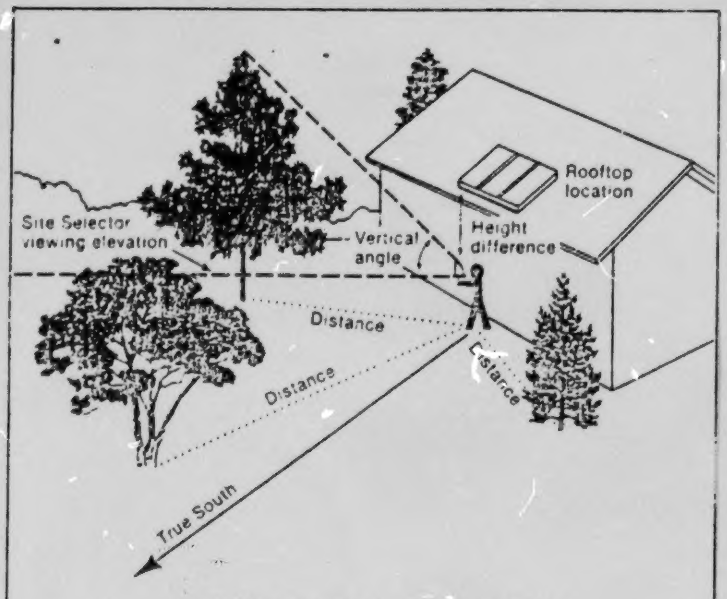
Permanent site records have also been made by photographing through the eyepiece. Both pocket instamatic and 35 mm cameras have been use. A 35 mm camera produces a relatively good picture by using a lens cap with a 1 mm hole drilled in the center. This increases the field of view to a workable range (i.e. 6" to infinity). Shutter speeds used were on the order of 1 second.

EXTRAPOLATING A GROUND LEVEL SIGHTING TO ROOFTOP LEVEL

If it is impractical to make a rooftop sighting to determine optimum collector placement, the Solar Site Selector may be used at ground level and the results extrapolated to rooftop level.

Although the procedure detailed below may seem cumbersome at first it is actually straight-forward and easy to follow. After two or three trials it will become quite simple.

1. Select a ground level viewing point difectly beneath the rooftop location (fig. 1) and measure or estimate the height difference between the Solar Site Selector viewing elevation and the rooftop location.
2. Now make a tracing of the ground level shading profile following the procedure described in this booklet.
3. Estimate or measure the horizontal distance from the Solar Site Selector to each object that might appear in the rooftop profile. Note each object's distance on the tracing.
4. Using the solar Site Selector and the angle markings along the solar noon line on the grid, observe the vertical angle to the top of each object. Note each angle on the tracing.
5. Determine the height of each object above the Solar Site Sleector's viewing level using the height table as follows. In the left column, find the vetical angle most closely approximating that noted for the object. In the upper row, find the distance most closely approximating that for the object. The approximate height of the object is found where these two columns intersect.


Rooftop location
Site Selector viewing elevation
Vertical angle
Height difference
Distance
Distance
Distance
True South

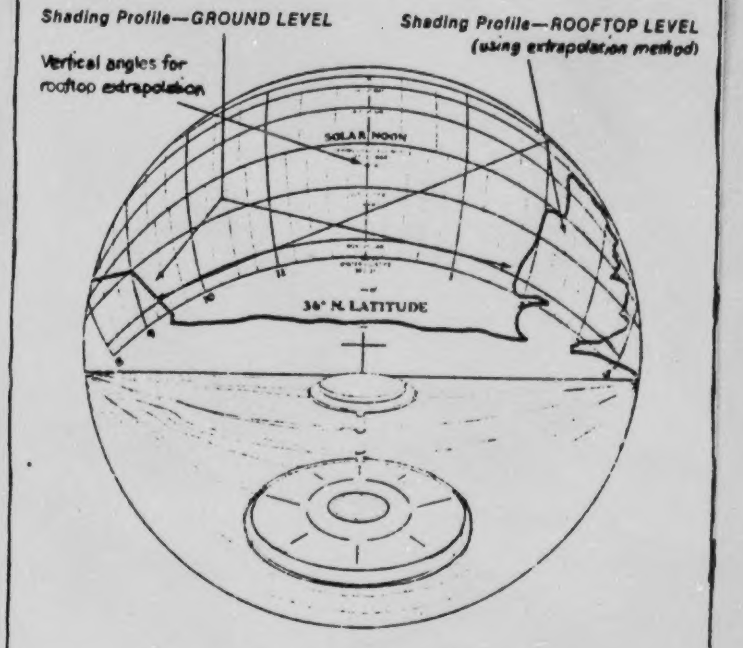
(Fig. 1) EXTRAPOLATION DIAGRAM

6. For each shading object subtract the height difference obtained in step 1 from the height just obtained in step 5. The difference is the vertical height of the object above the rooftop location
7. Returning to the height table in the same distance column find the figure that most closely approximates this new height. Now read back horizontally in the table to obtain the new angle. This is the angle that would actually be observed from a rooftop sighting.
8. Plot the new vertical angles for each shading object on the tracing. The rooftop shading profile can now be drawn below the ground level profile (See fig. 2).
9. All audit calculations for the rooftop position can now be computed in the manner described in Sighting and Calculating for Audits. NOTE: For greater precision, interpolate the actual figures on the height chart rather than using the closest approximation.

V-5-13

HEIGHT TABLE

Incline Angle	10	20	30	40	50	60	70	80	90
15°	3	5	8	11	13	16	19	21	24
20	4	7	11	15	18	22	25	29	33
25	5	9	14	19	23	28	33	37	42
30	6	12	17	23	29	35	40	46	52
35	7	14	21	28	35	42	49	56	63
40	8	17	25	34	42	50	59	67	76
45	10	20	30	40	50	60	70	80	90
50	12	24	36	48	60	72	83	95	107
55	14	29	43	57	71	86	100	114	129
60	17	35	52	69	87	104	121	139	156
65	21	43	64	86	107	129	150	172	193
70	27	55	82	110	137	165	192	220	247
75	37	75	112	149	187	224	261	299	335
80	57	113	170	227	284	340	397	454	510


Shading Profile—GROUND LEVEL
Shading Profile—ROOFTOP LEVEL (using extrapolation method)
Vertical angles for rooftop extrapolation
SOLAR NOON
36° N. LATITUDE

(FIG. 2) SHADING PROFILE DIAGRAM

MISCELLANEOUS FACTS TO CONSIDER

Site selection, solar glazing and collector orientation are of prime importance for optimum exposure to solar radiation. Many climatic conditions infringe on total clear day insolation (the amount of solar radiation striking a surface). However, two factors are crucial to maximum solar exposure: length of day and absence of shadows, both solid and partial.

Length of day as measured by hours of solar radiation becomes more critical as the latitude increases. To be most efficient and cost effective, solar glazing should have 5 to 6 hours per day of unobstructed sunshine at the winter solstice. Much more solar radiation is absorbed between the hours of 10 a.m. and 8 p.m. than during any combination of four hours preceding or following. Therefore, it is crucial to avoid shading patterns in the four mid-day hours.

A deviation of up to 20 degrees either east or west of true south will not significantly alter the performance of most solar collectors or solar glazing. This may be done if incoming radiation will be obstructed because of adjacent buildings or trees, or if morning fog or haze interferes.

A shaded collection area can be worse than no collection area at all. If If 10% of the collection area is shaded the system's efficiency will drop by 20%; a 50% shaded collector will produce virtually no heat at all. Thus cost effectiveness is greatly reduced when there is a high shading factor.

Scattered deciduous trees can be beneficial since they drop their leaves for the winter season allowing partial radiation while providing shade in the summer. Evergreen trees may be selectively trimmed or removed.

The maximum amount of solar radiation is transmitted when the sun's rays are perpendicular to the absorbing surface. Optimum orientations achieved by finding the most ideal angle of incidence for as long a period as possible during the winter months. Although the winter solstice falls on Dec. 21st (when the sun reaches its lowest point), January is generally the coldest month of the year.

These facts apply to passive, hybrid, or active systems design, construction, or installation, both new and retrofit.

To determine the best location for new construction, a number of sightings should be taken. For retrofit where 20 feet or more of glazing is considered, a sighting at each end should be taken. Smaller areas should have one sighting taken from the center of the area being considered.

V-5-14

INSTRUMENT CARE

The Solar Site Selector has been designed to require minimum care. The base itself should be stored in the canvas bag as it is a sturdy but finely finished wood product. The winter grid should be stored flat or in the box as it was shipped. Sore the summer grid in its mailing tube loosely rolled in the direction it will be use.

If the unit is used in the rain on more than an occasional basis, it is recommended that the wood be varnished. The grid is produced on mylar which does not tear (like acetate) but does scratch. Canvas bag should be dry cleaned only.

CAUTION

If you are using the Solar Site Selector continually in direct bright sunlight you may wish to protect your eyes from glare by use of a filter (reflective film) applied directly to the exterior of the lens or to the entire surface of the grid.

Exercise extreme care when using the summer grid as its edges, while rounded, extend beyond the base and could cause damage if contact is made directly to the eye.

FOR FURTHER INFORMATION

AIA Research Corporation, Solar Dwelling Design Concepts. U.S. Department of Housing and Urban Development, 1976, 146 pp. (available from U.S. Government Printing Office, Washington, D.C. 20402).

Anderson, Bruce and Riordan, Michael. The Solar Home Book. Heating, Cooling and Designing with the Sun. Chesire Books. Harrisville, New Hampshire, 297 pp.

Balcomb, J. Douglas, et al. Passive Solar Design Handbook, Volume Two of Two Volumes: Passive Solar Design Analysis. U.S. Department of Energy, Washington, D.C. 20585. 270 pp.

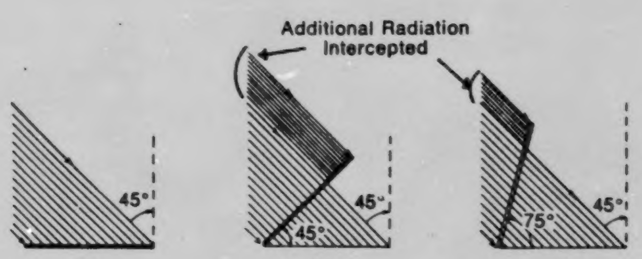
Barnaby, Charles S., et al. Solar for Your Present Home. San Francisco Bay Area Edition. California Energy Commission. Sacramento, n.d., 162 pp.

California Energy Commission. Solar Access: a Local Responsibility. 1111 Howe Ave., Sacramento, Ca. 95825, 1978, 35 pp.

Erley Duncan, et al. Site Planning for Solar Access: A Guidebook for Residential Developers and Site Planners. American Planning Association. Chicago, 1979, 149 pp. (available from U.S. Government Printing Office, Washington, D.C. 20402).

Jaffe, Martin, et al. Protecting Solar Access for Residential Development: A Guidebook for Planning Officials. The American Planning Association, Chicago, 1979, 154 pp. (available from U.S. Government Printing Office, Washington D.C. 20402).

It is advantageous to tilt the solar collector so that it is perpendicular to the sun's rays. Figure 1-10 further illustrates this advantage by showing the increase in energy intercepted when a collector is tilted from the horizontal. The optimum tilt occurs when the angle of the collector is the same as the incoming radiation. The maximum energy would be intercepted if the collector were to track the sun across the sky, but tracking collectors are very costly and bulky for home installation.



(a)
Collector Tilt Angle 0°

(b)
Collector Tilt Angle 45°

(c)
Collector Tilt Angle 75°

Fig. 1-10: Effect of tilting the collector on energy intercepted.

The rule generally followed for the tilt of the collector in the northern hemisphere is to face the collector to the south. The angle of the tilt is latitude plus 15 degrees for heating or minus 15° for cooling. For example, Woonsocket, Rhode Island, is located directly on latitude 42° N. If a collector were positioned for heating, its angle would be 57° (42° + 15° = 57°). If positioned for cooling, the collector's angle would be 27° (42° - 15° = 27°). When the collector is to be used for both heating and cooling, a reasonable rule is to have the angle of the collector equal the latitude. Thus, for Woonsocket, Rhode Island, the angle of the collector used for both heating and cooling would be 42°.

Since maximum solar intensity occurs at noon when the sun is due south (in the northern hemisphere), a collector should face directly south. If building conditions make this impossible, a variation of ± 15 degrees can be tolerated without serious effect on the solar radiation collected. Keep in mind, however, that an orientation 15 degrees east of south will advance the time of peak collection one hour. A similar orientation 15 degrees west of south will delay the time of peak collection one hour. For example, if the collector location is partially shaded in the early morning, aiming the collector west of south would decrease the morning collection while increasing the better afternoon collection.

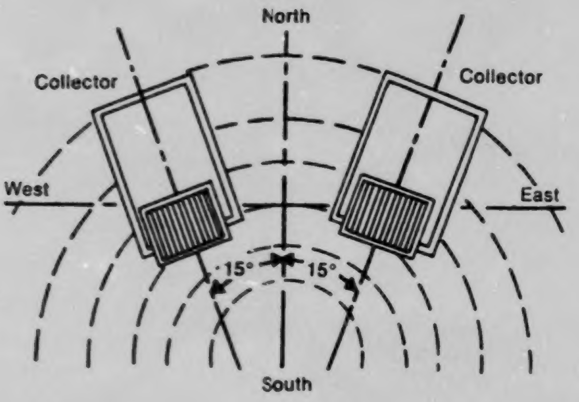


Fig. 1-11: Facing collectors other than due south has modest impact on energy collected. A 15 degree variation can be tolerated.

Ground mount collectors require consideration in the areas of potential shading, protection from vandalism and additional friction loss in the transport systems. Ground mounted collectors may be the best choice in sutuations where other limitations are difficult to overcome.

Placement of storage system often is the determining factor in the choice between liquid and air systems. Space availability often eliminates the air system as a choice. The structure, i. c., floor, supporting the storage unit must also be examined so that appropriate decisions can be made.

In addition to the above limitations that affect design decisions, local codes, property covenants, and zoning regulations may influence the design of the system.

Also, one should examine past energy demand patterns and occupant space usage; looking for potential conservation measures that could reduce the solar load.

When the building, site and orientation has been thoroughly examined, the collector decision must be made. Factors that affect this decision are:

A. Building load, demand periods, available options.

B. Energy use patterns, energy cost (present and projected future), and energy availability.

C. Solar application - Domestic hot water, space heating/cooling, commercial (launderies, agricultural, processing plants, etc.)

D. Reasonable solar fraction (percentage of total energy demand that can be provided by solar).

E. Physical and economic limitations.

One should compare the advantages and disadvantages of flat plate vs. concentrating collectors (active system). Once that decision is made, then compare liquid vs. air (for flat plate collectors) and make your decision based upon all the aforementioned parameters.

Factors affecting storage decisions are:

A. Location - inside or outside the building, distance from collector array and whether it will be above or below ground level.

B. Space requirements - Rock storage volume is much greater than liquid storage because of the difference in specific heat.

C. Container construction - Pressure requirements, steel, fiberglass, concrete, etc.? Which construction methods would be most applicable? If a liquid vessel, which would best serve the need, upright or horizontal tank?

D. Heat transfer medium - rock, water, water/glycol, phase change compounds.

E. Will it be accessible for maintenance and/or repairs?

F. Stratification of the storage medium temperatures.

When the preliminary decision on collector type, mounting method, storage type and storage container has been made, the subsystem components can be considered. Frequently the preliminary decisions are altered when this part of the process is considered in some detail. Example: an air system was chosen and storage space is no problem; however, when designing the required duct system with many dampers, it became apparent that a liquid system would be a better choice.

The following factors are those which affect decisions on subsystem components:

A. Piping - what size and configuration? How many and what type fittings, valves, access ports, temperature wells, etc.? How much insulation?

B. Ducts - what size and configuration? How many fittings, rectangular or round? How many dampers, manually or automatically controlled? What size and how many supply and return registers?

C. How many and what size heat exchangers will be required? Which type?

D. Pumps/Blowers - what size, which type and how many?

E. Control system - which type - electric, pneumatic, pneumatic-electric, solid-state? What is to be controlled - temperature, pressure, time, sequence?

F. Protective devices - for what purpose? Pressure relief, freeze protection, over current, over temperature? Air vents, filters, purge coils, expansion tanks, etc., all may play a very important role in system protection.

By this time you should be aware of the many decisions to be made in designing a solar system. Apprehension over what seems to be an insurmountable task is growing by "leaps and bounds". However, it is not as difficult as it may appear. We will undertake these tasks one at a time in the remaining modules of this course. Be alert and aware of the parameters of design as outlined herein and you will be successful.

SIZING, DESIGN AND RETROFIT

LOAD CALCULATION FACTORS AND PROCEDURES

STUDENT MATERIAL

for personal use in society.

The pattern of energy consumption may change in still other ways over the next several decades. Progressive electrification of energy usage is likely to continue. This is the best way to make use of nuclear energy, and improved technology can be expected to increase the efficiencies of electric power generation and application, so that electricity will be chosen more often over direct fuel combustion. There is a potential for limited use of solar power, primarily for supplying hot water and for comfort heating.

A possible future pattern for the flow of energy through the United States economy is shown in Figure 1-9C. Compared with the present, as shown previously, uranium and coal may provide more of the energy. The efficiency of conversion facilities may improve. Heat pumps, by drawing heat from the air, may augment the effectiveness of electrical heat. More efficient use of energy, as projected in Figure 1-9C, combined with a leveling off of per capita energy consumption and slower population growth, will tend to moderate the nation's overall energy consumption.

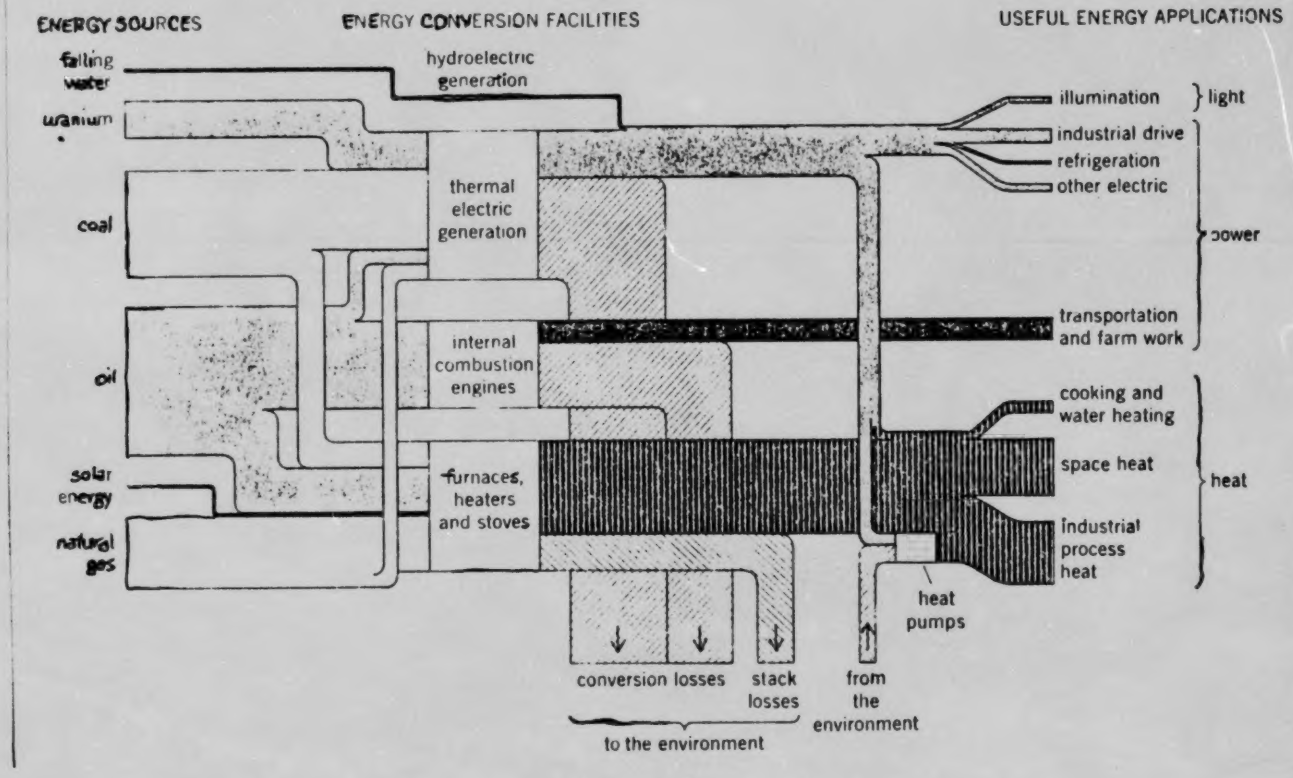

ENERGY SOURCES
ENERGY CONVERSION FACILITIES
USEFUL ENERGY APPLICATIONS
falling water
uranium
coal
oil
solar energy
natural gas
hydroelectric generation
thermal electric generation
internal combustion engines
furnaces, heaters and stoves
illumination
light
industrial drive
refrigeration
other electric
power
transportation and farm work
cooking and water heating
space heat
heat
industrial process heat
heat pumps
conversion losses
stack losses
to the environment
from the environment

Figure 1-9C.

SIZING, DESIGN AND RETROFIT

LOAD CALCULATION FACTORS AND PROCEDURES

Before the design process can begin one needs to make a thermal analysis of the building. A good foundation in load calculations is required before the analysis can be made with reasonable assurance of making logical decisions.

Recall, from the Energy Science I course, that there are three methods (modes) of heat transfer - conduction, convection and radiation. Heat load calculations must consider all three of these modes of transfer.

This module of instruction will give you a good feel for making load calculations using a variety of forms. The most popular load form in use today is the NESCA form; however, it is rather long and cumbersome for the beginner. Therefore, we will use the tabular form which has common elements to all forms and a short form similar to many of those used by manufacturers and utility companies.

The following procedures are those that will be followed in determining an initial load calculation, providing a foundation for heat gains and heat losses

Heat Loss

1. Select the outside design temperature (T_o) from table.

2. Select inside design temperature (T_i).

3. Design temperature difference (D.T.D.) for heating is T_i-T_o.

4. Calculate the net area (A) of walls, doors, windows, floors and ceilings for each different type of construction that is exposed to unconditioned space. Use the appropriate D.T.D. for partitions adjacent to unconditioned space but not exposed to outside conditions.

5. Calculate the U factor for each area in Step 4.

6. Calculate the transmission heat loss rate (Btu/hr.) for each living space; example: bedrooms, kitchen, living room, den, etc. $Q(Btu/hr.) = UA(T_i-T_o)$.

7. Total the transmission losses to the structure.

8. Calculate the infiltration losses (including ventilation losses if applicable).

9. Calculate the duct losses.

10. Add sum from line 8 and 9 to total from line 7 for total heat losses to the structure.

11. Subtract internal heat gains, where appropriate (usually insignificant in residences) from line 10 total. This figure represents the heat loss rate to be considered as the load.

12. Convert heat loss rate (Btu/hr.) to BTU/Degree Day.

Heat Gains

1. Select the outside design temperature (T_o) from table.

2. Select inside design temperature (T_i).

3. Design temperature difference (D.T.D.) for cooling is T_o-T_i.

4. Calculate the net area (A) of walls, doors, windows, floors and ceilings for each different type of construction that is exposed to undonditioned space. Use the appropriate D.T.D. for partitions adjacent to unconditioned space but not exposed to outside conditions.

5. Calculate the U factor for each area in Step 4.

6. Calculate the transmission heat gain rate for each living space. $Q(\text{Btu/hr.}) = UA(T_o - T_i)$.

7. Calculate the solar gain from windows, glass doors, walls, etc.

8. Calculate infiltration gains (include ventilation if appropriate).

9. Total figures from lines 6, 7, and 8.

10. Add gains from people, kitchen equipment, appliances and lighting to sum of line 9.

11. Add duct gain to total from line 10.

12. Multiply total from line 11 by 1.3 to allow for latent heat gains. Additionally add to this sum any unusual latent gain such as steam sources. This product represents the total cooling load - BTU/hr.

The above procedures are those that will be used in tabular calculations of heat gains/losses. The following information will serve as a basis for making these calculations.

HEAT LOSSES

Heat transmission losses, or more simply heat losses, from buildings may be divided into two groups: (1) the transmission losses through walls, floor, ceiling, glass and other surfaces and (2) the infiltration losses, or more correctly infiltration of cold air, through open doors and windows, cracks and crevices around doors and windows, which must be heated to the comfort level in the building.

HEAT TRANSMISSION THROUGH BUILDING SURFACES

Heat is transferred from warm room air to outdoor air by a three-step process. Heat is transferred from the room air to the inside surface of a wall or window, through the wall or window, and from the outside surface to the outdoor air. The rate of heat flow per unit of time from the building to the outdoors depends upon the surface area, A, an overall heat transfer coefficient, U, and the air temperature difference between the inside, T_i, and outside, T_o. Expressed in equation form:

$$Q = UA(T_i - T_o) \qquad (2\text{-}1)$$

where Q is heat flow rate, Btu/hr; A is wall area, ft^2; U is the overall heat transfer coefficient, Btu per $(hr)(ft^2)(^oF)$; T_i is indoor temperature, ^oF; and T_o is outdoor temperature, ^oF.

The overall heat transfer coefficient, often called the U factor, is determined by the reciprocal of the total thermal resistance, R_T, to heat flow:

$$U = \frac{1}{R_T} \qquad (2\text{-}2)$$

and

$$R_T = R_1 + R_2 + R_3 + R_4 + \text{etc.} \qquad (2\text{-}3)$$

where R_1, R_2, etc., are R factors, the individual resistances of the wall components.

The transfer of heat from the inside air to the wall is visualized as taking place through a thin film of air adjacent to the wall surface. This thin film has resistance, R_i, to heat flow determined by the film conductance, f_i,

$$R_i = \frac{1}{f_i} \tag{2-4}$$

and should be included in the determination of the overall U factor. Similarly, there is a thin film at the outside surface, the conductance of which, symbolized by f_o, is dependent upon the wind speed. The resistance of the outside film, R_o, is

$$R_o = \frac{1}{f_o} \tag{2-5}$$

During summer months, when the outside temperature is greater than the indoor temperature, heat is conducted into the building. The principles are the same as the foregoing, except that heat flow rate is determined

$$Q = UA\,(T_o - T_i) \tag{2-6}$$

where T_o and T_i have been interchanged from equation (2-1).

Surface conductances and resistances for air films for interior and exterior surfaces, for winter and summer, are tabulated in Table 2-1, at the end of this module. The winter values are based on wind velocity of 15 mph and summer values are based on wind velocity of 7 mph.

Dead air spaces between walls offer thermal resistance. The resistance values are tabulated in Table 2-2 for 3/4-inch and 4-inch spaces for winter and summer conditions. For spaces between 3/4 and 4 inches, values may be interpolated.

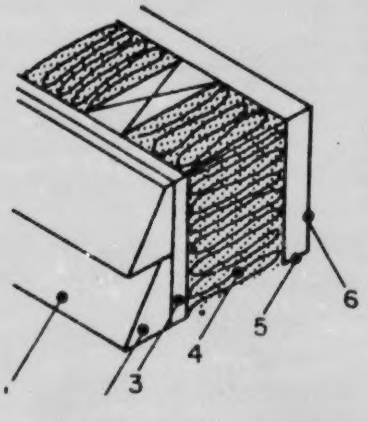
Resistance values for common building materials are tabulated in Table 2-3. U factors for windows and patio doors are tabulated in Table 2-4, and U factors for solid doors are listed in Table 2-5 with and without storm doors. The values in these tables correspond with more complete tables listed in ASHRAE Handbook of Fundamentals (1972).

TRANSMISSION COEFFICIENTS

The procedure for determining the overall heat transmission coefficients, U, for typical wall, roof, ceiling and floor construction is presented in this section. The values of R used are found in Tables 2-1 through 2-3. U factors for composite construction are determined in the following examples. U factors for other types of construction may be calculated by following these examples.

Example 1 - Frame Wall (2 x 4 studs)

	ITEM	R
1.	Outside film (15 mph wind, winter)	0.17
2.	Siding, wood 1/2 x 8 lapped)	0.81
3.	Sheathing (1/2 inch regular)	1.32
4.	Insulation batt (3-3 1/2 inch)	11.00
5.	Gypsum wall board (1/2 inch)	0.45
6.	Inside surface (winter)	0.68
	Total Resistance, R_T	14.43
	$U = 1/R_T$	0.07



The calculated U factor applies to the area between 2 x 4 studs. Because the resistance to heat flow through the 2 x 4 stud is different from the insulation, a correction is sometimes applied. However, the corrections usually amount to less than the accuracy of the R values. Corrections are therefore considered unnecessary.

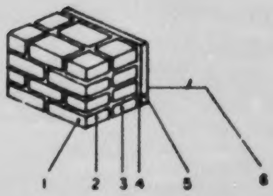
Example 2 - Frame Wall (2 x 6 studs)

From Example 1,	
R_T	14.43
Replace 3 1/2-inch insulation, subtract	11.00
	3.43
Add 5 1/2-inch insulation	19.00
New R_T	22.43
$U = 1/R_T$	0.04
Difference in U from Example 1	0.03
Percent Difference from 2 x 4 wall	43 percent

There is 43-percent reduction in heat loss for a 2 x 6 wall as compared with a 2 x 4 wall with correspondingly thicker insulation in the 2 x 6 wall.

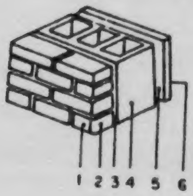
Example 3 - Solid Masonry Wall

	ITEM	R
1.	Outside film (15 mph wind, winter)	0.17
2.	Face brick (4 inch)	0.44
3.	Common brick (4 inch)	0.80
4.	Air space (3/4 inch)	1.28
5.	Gypsum board (1/2 inch)	0.45
6.	Inside surface	0.68
	Total resistance, R_T	3.82
	$U = 1/R_T$	0.26



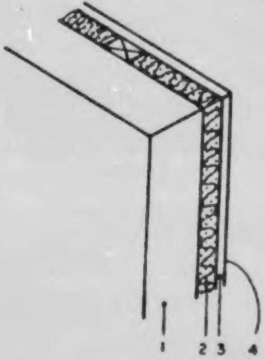
Example 4 - Masonry Walls

	ITEM	R
1.	Outside surface (15 mph)	0.17
2.	Face brick (4 inch)	0.44
3.	Cement mortar (1/2 inch)	0.10
4.	Cinder block (8 inch)	1.72
5.	Air space (3/4 inch)	1.28
6.	Gypsum board (1/2 inch)	0.45
7.	Inside surface	0.68
	Total Resistance, R_T	4.84
	$U = 1/R_T$	0.21



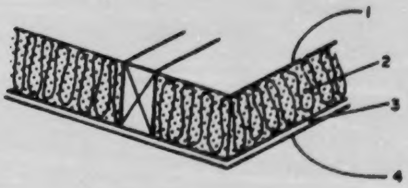
Example 5 - Basement Wall

	ITEM	R
1.	Concrete wall (8 inch)	0.64
2.	Insulation batt (2 inch)	7.00
3.	Gypsum board (1/2 inch)	0.45
4.	Inside surface	0.68
	Total Resistance, R_T	8.77
	$U = 1/R_T$	0.11



Example 6 - Insulated Ceiling, 6 inches

ITEM	R
1. Inside surface	0.68
2. Insulation batt (6 inch)	19.00
3. Gypsum board (1/2 inch)	0.45
4. Inside surface	0.68
Total Resistance, R_T	20.81
$U = 1/R_T$	0.05



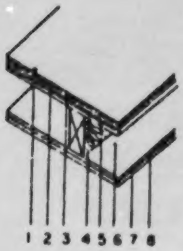
Example 7 - Insulated Ceiling, 9 inches

ITEM	R
1. Inside surface	0.61
2. Insulation (9 inch)	24.00
3. Gypsum board (1/2 inch)	0.45
4. Inside surface	0.61
Total Resistance, R_T	25.67
$U = 1/R_T$	0.04

Percent decrease of U with 9-inch insulation over 6-inch insulation, 20 percent

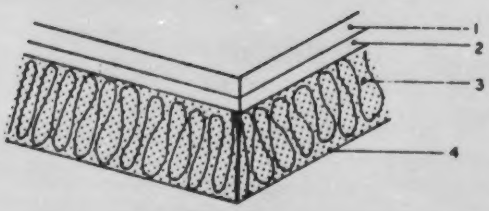
Example 8 - Floor

	ITEM	R
1.	Top surface	0.61
2.	Linoleum or tile	0.05
3.	Felt	0.06
4.	Plywood (5/8-inch)	0.78
5.	Wood subfloor (3/4-inch)	0.94
6.	Air space	0.85
7.	Acoustic ceiling tile (3/4-inch)	1.89
8.	Surface	0.61
	Total Resistance, R_T	5.79
	$U = 1/R_T$	0.17



Example 9 - Floor

	ITEM	R
1.	Carpet and fibrous pad	2.08
2.	Plywood (3/4-inch)	0.93
3.	Insulation (9-inch)	24.00
4.	Surface (still air)	0.61
	Total Resistance, R_T	27.62
	$U = 1/R_T$	0.04

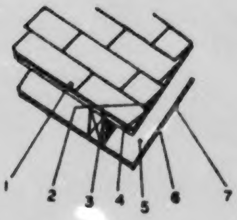


Example 10 - Basement

The heat loss from a heated basement should be based on a heat transfer coefficient for both wall and floor of U - 0.10. The temperature adjacent to basement walls and floor varies with the rate of heat transfer through the walls. The more heat that flows through the walls, the warmer will be the ground temperature. Below basement floors, a ground temperature equal to the ground water temperature is sometimes used. A temperature of 45° F is recommended as a rule-of-thumb for this course. If conditions warrant, a different temperature may be used.

Example 11 - Pitched Roofs (Heat Flow Up)

ITEM	R
1. Outside surface (15 mph)	0.17
2. Asphalt shingle roofing	0.44
3. Building paper	0.06
4. Plywood deck (5/8-inch)	0.78
5. Inside surface	0.61
Total Resistance, R_T	2.06
$U = 1/R_T$	0.49



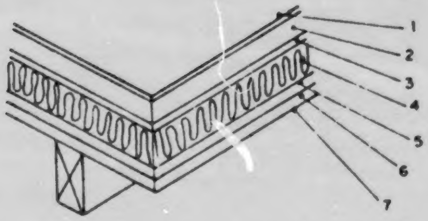
Example 12 - Pitched Roof with Air Space and Sheathing (Heat Flow Up)

See Example 11.

	ITEM	R
1.	Outside surface	0.17
2.	Wood shingle	0.94
3.	15 pound felt	0.06
4.	Plywood deck (5/8-inch)	0.78
5.	Air space	1.00
6.	Gypsum (1/2 inch)	0.45
7.	Inside surface	0.61
	Total Resistance, R_T	4.01
	$U = 1/R_T$	0.25

Example 13 - Pitched Roof with Mounted Collector (At Night)

	ITEM	R
1.	Outside surface	0.17
2.	Glass	1.13
3.	Air space (3/4-inch)	1.75
4.	Insulation	7.00
5.	15 pound felt	0.06
6.	Plywood (3/4-inch)	0.93
7.	Inside surface	0.61
	Total Resistance, R_T	11.65
	$U = 1/R_T$	0.09



HEAT LOSS BY INFILTRATION

Calculation of infiltration losses can be very complex. Experience and judgment are important to provide reasonable estimates. Of two methods used for calculating infiltration rates, only the simpler air change method is discussed in this module. Readers are referred to the ASHRAE Handbook of Fundamentals for details of the "Crack" method, which is probably more accurate. In either method, the objective is to determine the amount of heat required to raise the temperature of cold air which enters a building through cracks, open windows, and doors.

The volume of cold air expected to enter a room through cracks during a one-hour period depends on such factors as wind direction and speed, pressure differences in and outside the building, storm windows, air locks on outdoor entrances, and whether room doors are closed. The entering volume of cold air is expressed in terms of air changes per hour in the room under consideration. It is normally expected that storm doors and windows, or tight-fitting double-glazed windows will soon be widely adopted in new construction, particularly for solar heated and cooled houses. The average air changes for rooms with various fenestrations listed in Table 2-6 are in accordance with Chapter 19, ASHRAE Handbook of Fundamentals (1972).

From the air change rate, per hour, the volume rate of air change per hour, V, is determined from the room volume. The heat loss from infiltration is calculated from

$$Q = 0.018\ V\ (T_i - T_o) \qquad (2\text{-}7)$$

where V is the volume change per hour; Q is Btu per hour.

When moisture is added to the air to maintain winter comfort conditions, heat will be required to evaporate the water vapor added to the building air. The rate of heat added is most conveniently calculated from the equation below:

$$Q = 79.5\ V\ (W_1 - W_o) \qquad (2\text{-}8)$$

where V is the infiltration rate, cfh; W_1 is humidity ratio of indoor air, dimensionless; W_o is humidity ratio of outdoor air, dimensionless.

Infiltration occurs primarily because of wind impacting on the building from a given direction. Therefore, only the rooms on one side of the building would be affected at a given time. The values in Table 2-6 account for this factor.

HEAT LOSS CALCULATION

Procedure

1. Select the design outdoor temperature (T_o).
2. Select the indoor design temperature (T_1) at 68° F. (If zone controls or clock thermostats are used to lower the temperature of unused rooms and at night, consideration should be given to selecting other indoor temperatures for specific periods of time.)
3. Determine net areas, A, of walls, roof, ceiling, windows, doors, and floor for each different type of construction.
4. Select U factors from Examples 1 through 13, or calculate appropriate U factors for specific wall type.
5. Calculate heat transmission loss rate from:

   $$Q = UA\ (T_1 - T_o) \qquad (2\text{-}1)$$

   through each type of surface.
6. Sum the transmission losses.
7. Determine infiltration losses.
8. Add the infiltration losses to the transmission losses to obtain the total heat loss from the building.
9. Determine the design heat loss rate for the building for each degree day.

Temperatures of Unheated Spaces

Attic Temperature - The attic temperature is determined from a balance of heat flow into and out of the attic. Heat flow into the attic is from the ceiling; heat flow out is through the roof surfaces and end walls. The general formula for determining attic temperature is:

$$T_{at} = \frac{A_cU_cT_c + T_o(A_rU_r + A_wU_w)}{A_cU_c + A_rU_r + A_wU_w} \qquad (2\text{-}9)$$

where

T_{at}	is attic temperature, °F
T_c	is room temperature, °F
T_o	is outside temperature, °F
A_c	is ceiling area, ft^2
A_r	is roof area, ft^2
A_w	is roof wall area, ft^2
U_c	is ceiling U factor, $Btu/(hr)(ft^2)(°F)$
U_r	is roof U factor, $Btu/(hr)(ft^2)(°F)$
U_w	is wall U factor, $Btu/(hr)(ft^2)(°F)$

Example 14 - Attic Temperature for a Wood Shingled Roof

Calculate attic temperature for a wood shingled roof with the given dimensions. T_o is -9° F, T_c is 68° F. See Example 6 for ceiling U factor, $U_c = 0.05$. See Example 12 for roof U factor, $U_r = 0.25$. For Example 1, for no insulation and 3 1/2-inch air space, U factor for wall is:

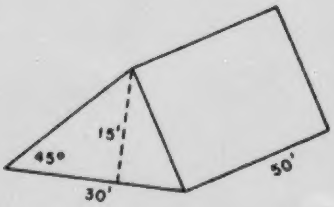
R_T from Example 1	14.43
Subtract insulation	-11.00
Subtract gypsum board	- 0.45
Total Resistance, R_T	2.98
$U_w = 1/R_T$	0.34

Calculate Area:

$A_c = 30 \times 50 = 1500\ ft^2$

$A_r = \sqrt{2} \times 15 \times 50 \times 2 = 2120\ ft^2$

$A_w = 30 \times 15 \times 1/2 \times 2 = 450$



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$$T_{at} = \frac{(1500)(0.05)(68) + (-9)\,[(2120)(0.25) + (450)(0.34)]}{(1500)(0.5) + (2120)(.25) + (450)(0.34)}$$

$$T_{at} = \frac{5100 - 6147}{75 + 530 + 153} = -1.4^{\circ}\ F$$

Example 15 - Attic Temperature with Mounted Collector

Calculate the attic temperature with a collector mounted on one side of roof. From Example 13, U_r with collector is 0.09. A_rU_r in equation (2-9) consists of two parts:

A_r (with collector) = 1060 ft^2
A_r (without collector) = 1060 ft^2
U_r (with collector) = 0.09
U_r (without collector) = 0.25
$A_rU_r = (1060)(0.9) + (1060)(.25) = 360$

For Example 2-14,

$$T_{at} = \frac{(1500)(0.5)(68) + (-9)\,[360 + (450)(.34)]}{(1500)(0.5) + 360 + 153}$$

$$T_{at} = \frac{5100 - 4617}{588} = 0.8^{\circ}\ F$$

When ventilation is provided, at 0.5 cfm per square foot of ceiling, the attic temperatures must be reduced from those calculated in Examples 14 and 15. Thus, the attic temperature approaches outdoor temperature. Attic temperature may be assumed to be the outdoor temperature with well-insulated ceilings without significant error in heat loss calculation.

Unheated Garage - With similar detailed calculations, the temperature in any unheated garage may be calculated. For ease of calculation of heat losses, the garage temperature may be assumed to be the mean of the indoor and outdoor temperatures, thus:

$$T_G = \frac{T_o + T_i}{2} \qquad (2\text{-}10)$$

Example: With outdoor temperature of -9 °F, indoor temperature of 68° F, the garage temperature is:

$$T_G = \frac{(-9) + 68}{2} = 30^{\circ}\ F$$

Btu/Degree Day

The heat load based on degree days is determined as follows:

$$\frac{\text{Heat loss}}{\text{Design Temperature Diff.}} \times 24$$

An example of heat load based on degree days (DD) could be [68 -(-9)].

$$\frac{53{,}215 \times 24}{68 - (-9)} = 16590 \ \frac{\text{Btu}}{\text{DD}}$$

It is interesting to note that the overall U factor for the house for the above grade living space based on the computations in Figure 2-3 is

$$U_o = \frac{(53215 - 7921)}{2078 \times 77} = 0.28 \ \text{Btu/(hr)(ft}^2\text{)(}^o\text{F)}$$

where 7921 is the basement load and 2078 is the ft^2 living space above grade.

For the entire house including the basement,

$$U_o = \frac{53{,}215}{3260 \times 77} = 0.21 \ \text{Btu/(hr)(ft}^2\text{)(}^o\text{F)}$$

HEAT GAINS

Heat transmission into a building takes place by radiation and conduction from building surfaces and by infiltration of warm air into conditioned space. The detailed procedure is quite complex, taking into account the thermal and optical properties of the building materials, time of day, day of the year, solar radiation intensity, etc. The procedure described in this module is based on a simplified method using a design equivalent temperature difference.

Heat gain is computed by:

$$Q = UA(DTD) \qquad (2\text{-}11)$$

where

Q is rate of heat gain, Btu/hr.

A is area of surface, ft^2

U is heat transmission coefficient, Btu/(hr)(ft^2)(^oF)

DTD is design equivalent temperature difference.

The DTD for three design outdoor temperatures are listed in Table 2-7. U factors for typical construction may be computed in the manner shown in Examples 1 through 13. Heat gain through windows depends upon exposure to solar radiation; therefore, heat gains will differ for different window orientations. Heat gains directly in terms of Btu/(hr)(ft^2) are listed in Table 2-8. No credit is given for shade line below an overhang in the table. When a permanent overhang is provided, the shaded window is treated as a north-facing window. Average shade lines below an overhang for various latitudes and window orientation are given in Table 2-9. The overhang width multiplied by the shade factor determines the average effective shadow lines below the level of the overhand. Data are for August 1, averaged over 5 hours.

INFILTRATION

Infiltration in the summer is less than in winter because the temperature difference and wind velocity are less. Air changes per hour for the summer are listed in Table 2-6. Sensible heat gain is determined by equation (2-7) and latent heat gain by equation (2-8). Residential cooling loads are almost always based on sensible heat gains.

OCCUPANCY

Heat gains from human beings in a residence is usually assumed to be about 200 to 250 Btu per hour. For normally equipped kitchens, heat gain from appliances is assumed to be 1200 Btu per hour for determining cooling loads.

SOLAR EQUIPMENT

Heat gains from solar equipment in a residence, i.e., motors, heated pipes and ducts, will add to the cooling load. The heat gain could be significant from water storage tanks if the equipment room is not vented. While there are as yet insufficient data from solar heated and cooled houses to provide design tables, a heat gain equivalent to the kitchen load, 1200 Btu per hour, may be assumed.

LATENT HEAT

Latent heat load of 30 percent of the sensible heat load may be used.

COOLING LOAD

The differences between heat gains and cooling loads are important in calculating residential cooling loads. The cooling loads in residential buildings are primarily due to sensible heat flow and not to internal heat gains. It must be remembered that only a few days each season are design days, and a partial load condition exists for many hours during a season. Thus, an oversized system does not perform effectively with short term or intermittent operating cycles. Equipment should be of the smallest possible capacity and designed to operate for 24 hours a day, using the thermal storage available in interior walls, and furnishings, to reduce temperature excursions in the building.

PROCEDURE FOR CALCULATION

1. Determine the design outdoor summer temperature.
2. Establish an indoor design temperature (usually 75° F).
3. Determine net areas of building sub-structures.
4. Select U factors from Examples 1 through 13, or calculate U factor from appropriate tables.
5. Select the Design Equivalent Temperature Difference (DETD) from Table 2-7.

6. For windows, use heat gain rates given in Table 2-8 corrected for shading factors given in Table 2-9.
7. Calculate the sensible heat gain from conduction and radiation, using equation (2-11).

$$Q = UA(DETD) \tag{2-11}$$

8. Calculate the sensible heat gain due to infiltration, using Table 2-6.
9. Add heat gain from occupants and fixed appliances.
10. Sum the sensible heat gains.
11. Add 30 percent for latent cooling load.
12. Total the latent load and sensible heat gains to determine the total cooling load.

The cooling load calculation procedure follows the same guidelines as that of heating load calculations.

The tables, figs. 2-3 and 2-4 on the following pages, will give an example of the worksheet procedure.

Figure 2-3

HEATING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS	TOTALS
BEDROOM 1						
South wall	(15+3)x8	120	.07	77	647	
East wall	13.5x8	108	.07	77	282	
Windows (2)	3x4	24	.50	77	924	
Infiltration	2/3x15x13.6x8	1088	.018	77	1508	3361
BEDROOM 2						
East wall	14x8	112	.07	77	604	
North wall	11x8	76	.07	77	410	
Window	3x4	12	.50	77	462	
Infiltration	2/3x11x11x8	645	.018	77	894	2370
BATHROOM						
North wall	8x8	60	.07	77	323	
Window	2x2	4	.50	77	154	
Infiltration	3/4x7.5x11x8	495	.018	77	686	1163
BEDROOM 3						
North wall	12x8	84	.07	77	453	
West wall	10x8	80	.07	77	431	
Window	3x4	12	.50	77	462	
Infiltration	2/3x10x12x8	640	.018	77	887	2233
BEDROOM 4 & HALLWAY						
West wall	16x8	128	.07	77	690	
South wall	14x8	88	.07	77	474	
Window	2x3x4	24	.50	77	924	
Infiltration	2/3x14x16x8	1195	.018	77	1656	3744
LIVING ROOM						
South wall	32x8	203	.07	77	1094	
Door	3x7	21	.26	77	420	
Window	4x8	32	.62	77	1528	
East wall	13.5x8	108	.07	77	582	
Infiltration	2/3x19x13.5x8	1368	.018	77	1896	5520
DINING ROOM						
East wall	13.5x8	108	.07	77	582	
North wall	11x8	88	.07	77	474	
Infiltration	1/3x11x13.5x8	396	.018	77	549	1605

Figure 2-3 (continued)

HEATING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U COEFF.	TEMP. DIFF. [68-(-9)]	HEAT LOSS	TOTALS
KITCHEN, BREAKFAST						
North wall	18x8	122	.07	77	657	
Window	2.5x4	10	.50	77	385	
Window	3.4	12	.50	77	462	
Infiltration	1x18x11x8	1584	.018	77	2195	3699
FAMILY ROOM						
North wall	21.5x8	136	.07	77	733	
Patio door	6x6	36	.58	77	1608	
West wall	13x8	104	.20	77	1602	
South wall	22x8	176	.52	38	3478	
Infiltration	2x13x22x8	4576	.018	77	6342	13763
HALL						
West wall	17x8	136	.52	38	2687	
Infiltration	1x8x8x17	1088	.018	77	1508	4195
BASEMENT						
North wall	54x8	432	.10	23	994	
West wall	28x8	224	.10	23	515	
South wall	54x8	432	.10	23	994	
East wall	28x8	224	.10	23	515	
Floor	32x28	896	.10	23	2061	
Floor	13x22	286	.10	23	658	
Infiltration	1/6x54x13x8+ 1/6x15x32x8	1576	.018	77	2184	7921
CEILING						
Second floor	32x28	896	.04	77	2760	
Family room	13x22	286	.04	77	681	3641

TOTAL 53215

Figure 2-4

COOLING WORKSHEET for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DDT	HEAT GAIN	TOTALS
BEDROOM 1						
South wall	18x8	120	.07	19	160	
East wall	13.5x8	108	.07	19	144	
Windows south	3x4	24	27		648	
Infiltration	1632	816	.018	14	205	1157
BEDROOM 2						
East wall	14x8	112	.07	19	149	
North wall	11x8	76	.07	19	101	
Window	3x4	12	27		324	
Infiltration	968	484	.018	14	122	696
BATHROOM						
North wall	8x8	60	.07	19	80	
Window	2x2	4	27		108	
Infiltration	660	660	.018	14	166	354
BEDROOM 3						
North wall	12x8	84	.07	19	112	
West wall	10x8	80	.07	19	106	
Window	3x4	12	27		324	
Infiltration	960	480	.018	14	121	663
BEDROOM 4 and HALLWAY						
West wall	16x8	128	.07	19	170	
South wall	14x8	88	.07	19	117	
Window	3x4	24	27		648	
Infiltration	1792	896	.018	14	226	1161
LIVING ROOM						
South wall	32x8	203	.07	19	270	
Door	3x7	21	.47	19	188	
Window	4x8	32	21		672	
East wall	13.5x8	108	.15	11	178	
Infiltration	2052	1026	.018	14	258	1566
DINING ROOM						
East wall	13.5x8	108	.07	19	144	
North wall	11x8	88	.07	19	117	
Infiltration	1188	198	.018	14	50	311

Figure 2-4 (continued)

COOLING WORKSHEET
for Example Building

BUILDING SECTION	SIZE OR VOLUME	NET AREA OR VOLUME	U or UNIT HEAT GAIN	DDT	HEAT GAIN	TOTALS
KITCHEN, BREAKFAST						
North wall	1848	122	.07	19	162	
Windows		22	27		594	
Infiltration	1584	1584	.918	14	399	1155
FAMILY ROOM						
North wall	21.5x8	136	.07	19	180	
West wall	13x8	104	.20	19	395	
South wall	22x8	176	.52	7	640	
Patio door	6x6	36	21		756	
Infiltration	2288	2288	.018	14	577	2548
HALL						
West wall	17x8	136	.52	7	495	
Infiltration	1088	1088	.018	14	274	769
CEILING						
Second floor	32x28	896	.04	39	1398	
Family room	13x22	286	.04	39	446	1844

TOTAL	12224
4 occupants x 225	900
Kitchen Appliances	1200
Total Sensible Heat Gain	14324
Latent Heat Gain (30%x14324)	4297
Latent + Sensible Heat Gain	18621
Cooling Load, Btu/hr	18621

No load is calculated for basement. No credit for cool basement taken.

Table 2-1. **Surface Conductances and Resistances for Air Films;**
Conductance-Btu/(hr)(ft^2)(^{o}F)
Resistance-(hr)(ft^2)(^{o}F)

ITEMS	WINTER		SUMMER	
	f_i	R_i	f_i	R_i
INTERIOR SURFACES				
Ceiling	1.63	0.61	1.08	0.92*
Sloped ceiling 45^o	1.60	0.62	1.32	0.76*
Walls and windows	1.46	0.68	1.46	0.68
Floor	1.08	0.92	1.08	0.92
EXTERIOR SURFACES				
Roofs, walls and windows	6.00	0.17+	4.00	0.25†

* Heat flow direction reversed from winter conditions

+ 15 mph wind

† 7.5 mph wind

Table 2-2. Resistance Values for Air Spaces (hr)(ft^2)(^{o}F)

ITEM \ Air Space	WINTER		SUMMER	
	3/4"	4"	3/4"	4"
Flat Roof	1.02	1.12	0.87	0.94
Wall	1.28	1.16	1.01	1.01

Table 2-3. Resistance Values for Building Materials $(hr)(ft^2)(^oF)/Btu$

TYPE AND MATERIAL	R	TYPE AND MATERIAL	R
BUILDING BOARD		SIDING	
Asbestos-cement: 1/8"	0.03	Asbestos-cement	0.21
1/4"	0.06	Wood shingles, 16"	0.87
Gypsum: 3/8"	0.32	Wood bevel, 1/2 x 8	0.81
1/2"	0.45	Wood bevel 3/4 x 10	1.05
Plywood: 1/4"	0.31	Wood plywood, 3/8	0.59
3/8"	0.47	Aluminum or steel	0.61
1/2"	0.62	Insulating Board:	
3/4"	0.93	3/8" normal	1.82
Insulating Board 25/32"	2.06	3/8" foiled	2.96
Regular 1/2"	1.32		
Laminated Paper 3/4"	1.50	FINISH FLOORING	
Acoustic Tile 1/2"	1.25		
3/4"	1.89	Carpet and fibrous pad	2.08
Hardboard 3/4"	0.92	Carpet and rubber pad	1.23
Particle Board 5/8"	0.82	Cork tile, 1/8"	0.28
Wood Subfloor 3/4"	0.94	Terrazzo, 1"	0.08
MASONRY		Tile, asphalt, linoleum, vinyl, rubber	0.05
		Hardwood	0.08
Concrete 6"	0.48		
8"	0.64	INSULATION	
10"	0.80		
Concrete Blocks,		Blanket and Batt: 2-2 3/4"	7.00
3 oval core		3-3 1/2"	11.00
Sand and Gravel 4"	0.71	5 1/4-6 1/2"	19.00
8"	1.11	Loose Fill	
12"	1.28	Cellulose, per inch	3.70
Cinder 4"	1.11	Sawdust, per inch	2.22
8"	1.72	Perlite, per inch	2.70
12"	1.89	Mineral fibre	
Lightweight 4"	1.50	(rock, slag, glass) 3"	9.00
8"	2.00	4 1/2"	13.00
12"	2.27	6 1/4"	19.00
Concrete Blocks,		7 1/2"	24.00
2 rect. core		Vermiculite, per inch	2.20
Sand and Gravel 8"	1.04		
Lightweight 8"	2.18	ROOFING	
Common Brick 2"	0.40		
4"	0.80	Asphalt	0.44
Face Brick 2"	0.22	Wood	0.94
4"	0.44	3/8" Built-up	0.33
		Woods: oak, maple per inch	0.91
BUILDING PAPER		fir, pine, softwoods	
		per inch	1.25
15# felt	0.06	3/4"	0.94

Table 2-4. U Factors for Windows and Patio Doors Btu/(hr)(ft^2)(^{o}F)

DESCRIPTION	WINTER
SINGLE GLASS	
Metal sash	1.13
Wood sash, 80% glass	0.02
DOUBLE GLASS:	
1/4" Air Space	
Metal sash	0.65
Wood sash, 80% glass	0.62
Wood sash, 60% glass	0.55
1/2" Air Space	
Metal sash	0.70
Wood sash, 80% glass	0.49
TRIPLE GLASS	
1/4" Air Space	
Metal sash	0.56
Wood sash, 80% glass	0.45
STORM WINDOWS	
1" to 4" Air Space	
Wood	0.50
Metal	0.56
SLIDING PATIO DOORS	
Single Glass	
Wood frame	1.07
Metal frame	1.13
Double Glass, 1/2" Air Space	
Wood frame	0.58
Metal	0.64

Table 2-5. U Factors for Solid Doors Btu/(hr)$(ft^2)(^oF)$

THICKNESS (IN)	WINTER			SUMMER WITHOUT STORM DOOR
	WITHOUT STORM DOOR	WITH STORM DOOR, 50% GLASS		
		WOOD	METAL	
1	0.64	.030	0.39	0.61
1 ¼	0.55	0.28	0.34	0.53
1 ½	0.49	0.27	0.33	0.47
2	0.43	0.24	0.29	0.42

Table 2-6. Air Changes for Average Residential Conditions.

KIND OF ROOM	AIR CHANGE PER HOUR	
	WINTER	SUMMER
Room with no windows or exterior doors	1/3	1/6
Rooms with windows or exterior doors on one side	2/3	1/2
Rooms with windows or exterior doors on two sides	1	2/3
Rooms with windows or exterior doors on three sides	1 1/3	1
Entrance halls and air locks	1 1/2	1

Table 2-7. Design Equivalent Temperature Differences (°F)

DESIGN OUTDOOR TEMPERATURE	85	95		105
TEMPERATURE RANGE DURING DAY	15-25	15-25	>25	>25
WALLS AND DOORS				
Wood frame and doors	14	24	19	29
Masonry	6	16	11	21
CEILINGS AND ROOF				
Under vented attic, dark roof	34	44	39	49
Built-up roof (no ceiling), light roof	26	36	31	41
FLOORS				
Over unconditioned rooms and open crawl space	5	15	10	20
Over basement, enclosed crawl space	0	0	0	0

Table 2-8. Design Heat Gains Through Windows Btu/(hr)(ft^2)

OUTDOOR DESIGN TEMPERATURE	SINGLE PANE			DOUBLE PANE		
	85	95	105	85	95	105
NO AWNINGS OR INSIDE SHADING						
North	23	31	38	19	24	28
Northeast; Northwest	56	64	71	46	51	55
East and West	81	89	96	68	73	77
Southeast; Southwest	70	78	85	59	64	68
South	40	48	55	33	38	42
WITH DRAPERIES OR VEN. BLINDS						
North	15	23	30	12	17	21
Northeast; Northwest	32	40	47	27	32	36
East and West	48	56	63	42	47	51
Southeast; Southwest	40	48	55	35	40	44
South	23	31	38	20	25	29
ROLLER SHADES, HALF DOWN						
North	18	26	33	15	20	24
Northeast; Northwest	40	48	55	38	43	47
East and West	61	69	76	54	59	63
Southeast; Southwest	52	60	67	46	51	55
South	29	37	44	26	32	36
AWNINGS						
North	20	28	35	13	18	22
Northeast; Northwest	21	29	36	14	19	23
East and West	22	30	37	14	19	23
Southeast; Southwest	21	29	36	14	19	23
South	21	28	35	13	18	22

Table 2-9. Shade Line Factors*
(5 hour average, 1 August)

WINDOW ORIENTATION	LATITUDE					
	25	30	35	40	45	50
East and West	0.8	0.8	0.8	0.8	0.8	0.8
Southeast; Southwest	1.9	1.6	1.4	1.3	1.1	1.0
South	10.1	5.4	3.6	2.6	2.0	1.7

* Multiply shade line factors by width of overhang to determine shadow line below overhang.

Once the tabular method is understood, the Short Form method is relatively easy, in that arbitrary decisions are easily made. The following short forms are typical examples of these forms.

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FORM 219R 5-76

Texas Power & Light Company

HEATING AND COOLING ESTIMATE
RESIDENTIAL

Customer ____________________

Address ____________________

Town ____________________ District ____________________

Prepared by: ____________________ Date ____________________

DESIGN CONDITIONS

COOLING D.B. °F	HEATING D.B. °F.	INSULATION	HOUSE FACES	OVERHANG OR SHADING
Outside Temp.	Inside Temp.	Ceiling	☐ North	
Inside Temp.	Outside Temp.	Wall	☐ East	
Temp. Diff.	Temp. Diff.	Floor	☐ South	
		Window	☐ West	

CONSTRUCTION	COOLING FACTOR							HEATING FACTOR	QUANTITY	COOLING BTUH	HEATING BTUH
1. **GLASS—SOLAR GAIN Windows—Doors** (Sq. Ft.: Note A)	No Shading	Inside Shading & Overhang 0′	1′	2′	3′	4′	Outside Shades				
North	32	14	0	0	0	0	0				
Northeast	50	23	21	19	19	17	17				
East	50	23	20	19	17	14	13				
Southeast	80	36	26	19	8	0	21				
South	100	45	33	10	0	0	26				
Southwest	155	70	51	39	17	0	42				
West	190	86	82	80	67	57	57				
Northwest	128	58	54	51	51	49	37				
Horizontal	185										

	Uo	DESIGN D.B. TEMP. DIFFERENTIAL 20	22	25	30	65	60	55	50	QUANTITY	COOLING BTUH	HEATING BTUH
2. **GLASS TRANSMISSION** (Sq. Ft.)		Uo x TD				Uo x TD						
Standard—Single Glazing	1.13	22.6	24.9	28.2	33.9	73.4	67.8	62.2	56.5			
Insulating—Double Glazing	.78	15.6	17.2	19.5	23.4	50.7	46.8	42.9	39.0			
Storm Window	.67	13.4	14.7	16.8	20.1	43.6	40.2	36.8	33.5			
Storm or Insulating Glass with Thermal Break	.56	11.2	12.3	14.0	16.8	36.4	33.6	30.8	28.0			
3. **DOORS**—(Sq. Ft.)		Uo x TD				Uo x TD						
Solid Wood or Hollow Core	.55	11.0	12.1	13.8	16.5	35.8	33.0	30.3	27.5			
Wood with Storm Door	.34	6.8	7.5	8.5	10.2	22.1	20.4	18.7	17.0			
Metal with 1½" Urethane	.11	2.2	2.4	2.8	3.3	7.2	6.7	6.1	5.6			
4a. **FLOOR: SLAB** (Linear Ft. Exposed Edge)		Uo x zero				Uo x TD						
No Edge Insulation	.81					52.6	48.6	44.6	40.5			
R-4 Edge Insulation	.68					44.2	40.8	37.4	34.0			
R-7 Edge Insulation	.55					35.8	33.0	30.2	27.5			
4b. **FLOORS: ENCLOSED CRAWL SPACE** (Sq. Ft.)		Uo x zero				Uo x (TD-20)						
No Insulation	.270					12.2	10.8	9.4	8.1			
R-7 Insulation	.093					4.2	3.7	3.2	2.8			
R-11 Insulation	.073					3.3	2.9	2.6	2.2			
R-13 Insulation	.060					2.7	2.4	2.1	1.8			
R-19 Insulation	.046					2.1	1.8	1.6	1.4			
R-22 Insulation	.039					1.8	1.6	1.4	1.2			
4c. **FLOORS: OPEN CRAWL SPACE** (Sq. Ft.)		Uo x (TD-5)				Uo x TD						
No Insulation	.374	5.6	6.4	7.5	9.4	24.3	22.4	20.6	18.7			
R-7 Insulation	.103	1.5	1.8	2.1	2.6	6.7	6.2	5.7	5.2			
R-11 Insulation	.073	1.1	1.2	1.5	1.8	4.7	4.4	4.0	3.6			
R-13 Insulation	.064	1.0	1.1	1.3	1.6	4.2	3.8	3.5	3.2			
R-19 Insulation	.046	0.7	0.8	0.9	1.2	3.0	2.8	2.5	2.3			
R-22 Insulation	.041	0.6	0.7	0.8	1.0	2.7	2.5	2.3	2.0			
5. **WALLS** (Sq. ft.) NET		Uo x TD				Uo x TD						
No Insulation—Solid Masonry	.389	7.8	8.6	9.7	11.7	25.3	23.3	21.4	19.5			
No Insulation—Wood Siding	.320	6.4	7.0	8.0	9.6	20.8	19.2	17.6	16.0			
No Insulation—Brick Veneer	.240	4.8	5.3	6.0	7.2	15.6	14.4	13.2	12.0			
R-5 Insulation	.128	2.6	2.8	3.2	3.8	8.3	7.7	7.0	6.4			
R-7 Insulation	.109	2.2	2.4	2.7	3.3	7.1	6.5	6.0	5.5			
R-11 Insulation	.075	1.5	1.7	1.9	2.3	4.9	4.5	4.1	3.8			
R-13 Insulation	.065	1.3	1.4	1.6	2.0	4.2	3.9	3.6	3.3			
R-16 Insulation	.054	1.1	1.2	1.4	1.6	3.5	3.2	3.0	2.7			
R-19 Insulation	.047	0.9	1.0	1.2	1.4	3.1	2.8	2.6	2.4			
R-24 Insulation	.038	0.8	0.8	1.0	1.1	2.5	2.3	2.1	1.9			

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CONSTRUCTION	Uo	COOLING FACTOR DESIGN D.B. TEMP. DIFFERENTIAL 20	22	25	30	HEATING FACTOR 65	60	55	50	QUANTITY	COOLING BTUH	HEATING BTUH
6a CEILINGS—WITH ATTIC (Sq. Ft.) (*Note B)		Uo x (TD + 40)				Uo x TD						
No Insulation	.598	35.9	37.1	38.9	41.9	38.9	35.9	32.9	29.9			
R -4 Insulation	.176	10.6	10.9	11.4	12.3	11.4	10.6	9.7	8.8			
R -7 Insulation	.114	6.8	7.1	7.4	8.0	7.4	6.8	6.3	5.7			
R-11 Insulation	.079	4.7	4.9	5.1	5.5	5.1	4.7	4.3	4.0			
R-19 Insulation	.048	2.9	3.0	3.1	3.4	3.1	2.9	2.6	2.4			
R-22 Insulation	.042	2.5	2.6	2.7	2.9	2.7	2.5	2.3	2.1			
R-26 Insulation	.036	2.2	2.2	2.3	2.5	2.3	2.2	2.0	1.8			
R-30 Insulation	.032	1.9	2.0	2.1	2.2	2.1	1.9	1.8	1.6			
R-33 Insulation	.029	1.7	1.8	1.9	2.0	1.9	1.7	1.6	1.5			
R-38 Insulation	.025	1.5	1.6	1.6	1.8	1.6	1.5	1.4	1.2			
6b. CEILINGS—NO ATTIC (Sq. Ft.) (*Note B)		Uo x (TD + 45)				Uo x TD						
No Insulation	.470	30.6	31.5	32.9	35.3	30.6	28.2	25.9	23.5			
R -4 Insulation	.160	10.4	10.7	11.2	12.0	10.4	9.6	8.8	8.0			
R -5 Insulation	.130	8.5	8.7	9.1	9.8	8.5	7.8	7.2	6.5			
R -7 Insulation	.109	7.1	7.3	7.6	8.2	7.1	6.5	6.0	5.5			
R-11 Insulation	.076	4.9	5.1	5.3	5.7	4.9	4.6	4.2	3.8			
R-19 Insulation	.047	3.1	3.1	3.3	3.5	3.1	2.8	2.6	2.4			
R-26 Insulation	.035	2.3	2.3	2.4	2.6	2.3	2.1	1.9	1.8			
R-30 Insulation	.031	2.0	2.1	2.2	2.3	2.0	1.9	1.7	1.6			
7. SENSIBLE INFILTRATION VOL. METHOD (Cu. Ft.) (*Note C)		Uo x TD				Uo x TD						
0.50 Well Above Standard	.009	0.18	0.20	0.22	0.27	0.58	0.54	0.50	0.45			
0.67 Above Standard	.012	0.24	0.26	0.30	0.36	0.78	0.72	0.66	0.60			
0.75 Standard	.013	0.26	0.29	0.32	0.39	0.84	0.78	0.72	0.65			
1.00 Average	.018	0.36	0.40	0.45	0.54	1.17	1.08	0.99	0.90			
1.50 Below Average	.027	0.54	0.59	0.68	0.81	1.76	1.62	1.48	1.35			
2.00 Well Below Average	.036	0.72	0.79	0.90	1.08	2.34	2.16	1.98	1.80			
8. PEOPLE—SENSIBLE ONLY (Avg. No.)		250										

	FACTOR	QUANTITY	COOLING BTUH	HEATING BTUH
9. **SUB TOTAL—SENSIBLE HEAT** (Total 1 through 8)				
		DUCT FACTOR (From Table below)	X	X
10. **TOTAL SENSIBLE—INCLUDING DUCT LOSS** (Multiply Line 9 Sub Total by Duct Factor)				
11. **INFILTRATION - LATENT HEAT** (Cu. Ft.)	FACTOR			
0.50 Well Above Standard	0.332			
0.67 Above Standard	0.444			
0.75 Standard	0.497			
1.00 Average	0.633			
1.50 Below Average	0.994			
2.00 Well Below Average	1.333			
12. **PEOPLE—LATENT HEAT** (Avg. No.)	200			
13. **TOTAL LOAD BTUH** (Total 10, 11, 12)				
14. **UNITARY COOLING EQUIPMENT REQUIREMENT** (BTUH): Line 10 x 1.25				
15. **INSTALLED EQUIPMENT CAPACITY** (BTUH) Use Larger of Line 13 or Line 14				
		(*Note G)		
			TONS	KW

COST OF OPERATION ESTIMATE FOR HEATING AND COOLING

COOLING (Line 13) MBTUH ÷ EER = KW X Hours = KWH/Season @ $ /KWH = $ /Season
Use 1700 Hrs. for 75 Maintained
Use 1400 Hrs. for 78 Maintained
(*Note D)

HEATING (Line 10) MBTUH X (*Note E) KWH/MBTU ÷ S.P.F. = KWH/Season @ $ /KWH = $ /Season
Use 1.0 if Furnace
Use 2.0 if Heat Pump
(*Note F)

Total $ /Year

DUCTWORK LOCATION*	INSUL. THICKNESS	COOLING FACTOR	HEATING
Attic—vented	1"	1.15	1.25
—vented	2"	1.10	1.15
—unvented	1"	1.20	1.20
—unvented	2"	1.15	1.10
Crawl space—vented	2"	1.10	1.15
—unvented	1"	1.05	1.10
[illegible]	1"	1.00	1.00

*Note A 1. Use only the one largest Solar Gain.
2. Multiply the Factors by .90 for Plate Glass .85 for double glass
3. No Solar Gain is used if overhang exceeds 4', or if glass is shaded by permanent structure
*Note B For light colored roof, Multiply COOLING Factor by .75
*Note C See infiltration Construction Definitions (See pad back)
*Note D Full load equivalent run hours
*Note E Annual Heating Consumption Factor from TP&L Service Area Weather Data (See pad back)
*Note F S.P.F. is Seasonal Performance Factor of Heating Equipment
*Note G 12,000 BTU/TON 3413 BTU/KW

INFILTRATION CONSTRUCTION DEFINITIONS
(AS RELATED TO E-OK PROGRAM)

METHOD OF REDUCING INFILTRATION OF UNCONDITIONED AIR
(Ranges from 0.67 to 1.75 Air Changes Per Hour)

INFILTRATION REDUCED BY:	E-OK POINTS	AIR CHANGES SAVED PER HOUR
A. Soleplate Sealed (33%)	13	0.3564
B. Wiring & Plumbing Holes and Furrdowns Sealed (26%)	10	0.2808
C. Exterior Doors & Windows Weather Stripped (7%)	3	0.0756
D. Exterior Door & Window Rough Openings Caulked (17%)	6	0.1836
E. Attic Access in Conditioned Space Weather Stripped (2%)	2	0.0216
F. Outside Sheathing Holes Sealed and Polyethylene Film Installed (7%)	3	0.0756
G. Ventless or Dampered Range Hood Installed (8%)	3	0.0864
TOTAL	40	1.0800

STRUCTURAL AIR CHANGES PER HOUR AS RELATED TO E-OK POINTS

AIR CHANGES PER HOUR	E-OK POINTS
0.50	40
0.67	40
0.75	36
1.00	27
1.25	18
1.50	9
1.75	0

TP&L SERVICE AREA WEATHER DATA

	TOWN	OUTSIDE TEMPERATURE DESIGN (DEG. F) **SUMMER 1%	2½%	WINTER	HEATING DEGREE DAYS BASE 65°	ANNUAL HEATING CONSUMPTION FACTOR IN KWH/MBTU FOR NOTED ROOM TEMPERATURE 70°	71°	72°	73°	74°	75°
CENTRAL DIVISION	Cleburne	102	100	10	2100	185	202	215	231	246	262
	Farmers Branch	101	99	10	2300	202	221	235	253	270	287
	Euless	102	100	10	2350	207	225	240	259	276	293
	Irving	102	100	10	2300	202	221	235	253	270	287
	Mesquite	101	99	10	2300	202	221	235	253	270	287
	Garland	101	99	10	2300	202	221	235	253	270	287
	Mineral Wells	102	100	10	2450	224	244	260	279	298	317
	Plano	101	99	10	2300	202	221	235	253	270	287
	Richardson	101	99	10	2300	202	221	235	253	270	287
	Waxahachie	101	99	10	2450	224	244	260	279	298	317
	Ennis	101	99	10	2450	224	244	260	279	298	317
	Lancaster	101	99	10	2300	202	221	235	253	270	287
	*Weatherford (Parker Co.)	102	100	10	2850	250	273	292	313	334	355
	*Ft. Worth (Tarrant Co.)	102	100	10	2350	207	225	240	259	276	293
	*Dallas (Dallas Co.)	101	99	10	2300	202	221	235	253	270	287
EASTERN DIVISION	Lufkin	98	96	15	2000	192	209	224	240	256	272
	Nacogdoches	98	96	15	2100	202	220	231	252	269	286
	Palestine	99	97	15	1900	182	202	212	228	243	258
	Crockett	99	97	15	2100	202	220	231	252	269	285
	Terrell	101	99	10	2850	250	273	292	313	334	355
	Tyler	99	97	15	2300	220	241	257	276	294	312
	Athens	100	98	15	2000	192	209	223	240	256	272
	*Centerville (Leon Co.)	100	98	15	1900	182	202	212	228	243	258
	*Jacksonville (Cherokee Co.)	99	97	15	2000	192	209	223	240	256	272
	*Wills Point (Van Zandt Co.)	100	98	10	2600	229	250	266	286	334	324
NORTHERN DIVISION	Decatur	101	99	10	2800	246	269	286	308	328	349
	Denison	101	99	10	2750	242	264	281	302	322	342
	Bonham	100	98	10	2700	238	259	276	297	317	336
	Gainesville	100	98	10	2400	211	230	246	264	281	299
	McKinney	100	98	10	2400	211	230	246	264	281	299
	Paris	100	98	10	2750	242	264	281	302	322	342
	Clarksville	100	98	10	2750	242	264	281	302	322	342
	Commerce	101	99	10	2850	250	273	292	313	334	355
	Sulphur Springs	101	99	10	2900	255	278	297	319	340	362
	Sherman	101	99	10	2500	220	240	256	275	293	312
	*Denton Exp. Sta. (Denton Co.)	102	100	10	2350	207	225	240	259	276	293
SOUTHERN DIVISION	Brownwood	102	100	10	2450	224	244	260	279	298	317
	Stephenville	102	100	10	2550	224	245	261	280	299	318
	Corsicana	102	100	15	2450	235	256	274	294	314	332
	Hillsboro	101	100	15	2200	210	230	246	264	282	299
	Taylor	101	99	20	1950	206	224	240	257	275	292
	Cameron	101	99	20	1600	169	184	197	211	225	239
	Temple	101	99	20	1850	195	213	228	244	260	277
	Belton	100	98	20	2000	211	230	246	264	281	299
	McGregor	100	98	20	2000	211	230	246	264	281	299
	Killeen	100	98	20	2000	211	230	246	264	281	299
	Waco	101	99	20	2000	211	230	246	264	281	299
	*Whitney (Bosque Co.)	101	99	15	2400	230	251	268	288	308	326

*City listed as typical for county

**The design data represents the highest temperature reached or exceeded for 1 and 2½ percent of all hours (2928) during the summer months of June through September. In a normal summer there would be approximately 30 hours at or above the 1% design and 73 at or above the 2½% design. ASHRAE Standards 90-75 recommends the 2½% design temperature.

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T. D. RESIDENTIAL LOAD FORM

OWNER OR BLDR:	ZONE:	TD#W
ADDRESS: PLAN#	AREA:	DATE:
		S'MAN:
DEALER:	FACING:	ENG'R:
REMARKS:		

SOURCE OF HEAT GAIN OR LOSS		"R" FACTORS	AREA OR QNTY	COOLING FACTORS	COOLING FACTORS OTHER	COOLING HEAT GAIN	HEATING FACTORS	HEATING FACTORS OTHER	HEATING HEAT LOSS
OUTSIDE DESIGN TEMPERATURE 75°I.D. - 78°W. B. - 50% R.H.				101°			15°		
CEILING	VENTED ATTIC			2.0			3.0		
	ROOF, NO ATTIC			1.8			2.2		
FLOORS	SLAB (PERIMETER)			0			50		
	P & B (SQ. FT.)			1.10			3.0		
GLASS WINDOWS & DOORS	W/DRAPES - BLINDS								
	NORTH OR SHADE			30					
	EAST-WEST			61					
	SOUTH			36					
	N.E.-N.W.			46					
	S.E.-S.W.			55					
	TOT. GLASS						80		
SKYLITES	PER FACING (OVER)			110			80		
CLEAR STORY	PER FACING (OVER)			70			80		
DOORS	WOOD-NO INSUL'N			16			145		
WALLS	GROSS AREA								
	LESS DOORS & WINDOWS								
	NET WALLS			4.0			4.0		
PEOPLE	(2) EA. BD RM			300					
KITCHEN	APPLIANCES			1200					
SENSIBLE HEAT GAIN & HEAT LOSS									
SENSIBLE & LATENT				1.30					
DUCT LOSS - TOTAL GAIN OR LOSS				1.10			1.10		
CAPACITY MULTIPLIER (COOLING)				1.11					
ESTIMATED LOAD (BTUH ÷ 12000)				TONS			BTUH / 3400	K.W.	
CAPACITY EQUIPTMENT SELECTED				TONS				K.W.	

FACTORS UNDER D.T. 101° BASED AS FOLLOWS:
CEILING: R19 - WALLS: R11 - GLASS: SINGLE PANE - P & B FLOOR: R-19

ALL FACTORS THIS LOAD FORM SELECTED FROM MANUEL "J" 1977
FOR OTHER DESIGN TEMPERATURES OR TYPE CONSTRUCTION SEE REVERSE
SIDE THIS SHEET, OR CONSULT MANUEL "J".

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ADDITIONAL FACTORS TO APPLY IN BLANK COLUMNS

SOURCE OF HEAT GAIN OR LOSS	▨	COOLING		HEATING
	INSULATION	FACTORS	FACTORS	FACTORS
DESIGN TEMPERATURE	▨	@100°	@105°	@15°
CEILINGS: UNDER UNCONDITIONED	NONE	11.5	11.6	36
SPACE OR VENTILATED	R11 - 3.0"	3.0	3.1	5.0
ATTIC	R13 - 3.6"	2.5	2.6	4.0
FOR COMBINATION	R19 - 6.0"	2.0	2.0	3.0
ROOF/CEILING ADD	R22 - 7.0"	1.7	1.8	2.5
0.3 to FACTOR LISTED	R26 - 8.0"	1.6	1.7	2.2
	R30 - 9.0"	1.4	1.5	1.9
	R33 -10.0"	1.3	1.4	1.7
	R44 -12.0"	1.0	1.1	1.3

		COOLING				HEATING	
GLASS: WINDOWS OR	▨	100° D. TEMP.		105° D. TEMP.		15° D. TEMP.	
DOORS WITH	FACING	SINGLE	DOUBLE	SINGLE	DOUBLE	SINGLE	DOUBLE
DRAPES OR BLINDS	N	30	20	30	20	80	50
HALF DRAWN.	NE/NW	45	35	50	35	"	"
	E/W	60	50	65	50	"	"
	SE/SW	55	45	55	45	"	"
	S	35	30	40	30	"	"

		COOLING				HEATING	
GLASS: CLEAR STORY,	▨	100° D. TEMP.		105° D. TEMP.		15° D. TEMP.	
NO SHADING.	FACING	SINGLE	DOUBLE	SINGLE	DOUBLE	SINGLE	DOUBLE
SKYLITES: FOR CLEAR GLASS	N	35	25	40	30	80	54
OR PLEXI., NO	NE/NW	70	55	70	55	"	"
SHADING, MULT.	E/W	95	75	95	80	"	"
FACTOR SHOWN	SE/SW	85	65	85	70	"	"
BY 1.50.	S	55	40	55	45	"	"

WALLS: WOOD FRAME,	R VALUE	COOLING		HEATING
WITH SHEETROCK,	▨	100° D. TEMP.	105° D. TEMP.	15° D. TEMP.
½" SHEATHING,	0	9.0	9.0	15
BRICK OR WOOD	R11-3.0"	4.0	4.0	4.0
VENEER.	R13-3.6"	3.5	4.0	4.0
	R19-6.0"	3.0	3.3	2.4
WALLS: AS ABOVE WITH 1"				
POLYSTYRENE IN	R13+R5	3.3	3.6	3.0
LIEU GYP. SHEATH'G				

NOTE: INFILTRATION HAS BEEN INCLUDED IN HTM FACTORS.

CAPACITY MULTIPLIERS: 3° TEMP. SWING @100° = 1.10
101° = 1.11
105° = 1.15

V-S-60

T.D. Apartment Load Form

Job Name:
Plan #:
TD#W:
Location:
Date:
S'MAN:
Dealer:
Apt. #:
Quan:
ENG'R:

Source of Heat Gain or Loss	Area or Quantity	Cooling Gain Factors		Cooling Gain Down	Cooling Gain Up	Heating Loss Factors		Heating Loss Down	Heating Loss Up
Outside Design Temperature		105	oth.	BTUH		15	oth	BTUH	
Ceiling: R-19 6" insulation		2				3			
Floor: Slab (Primeter)						50			
Floor: Other									
Glass: North		30							
East/West		65							
South		40							
NE & NW		50							
SE & SW		55							
Other Glass									
Total Glass						80			
Doors (Wood)		16				145			
Gross Exp. Wall									
Less Doors & Window									
Net Exp. Wall		4				4			
People (2 ea. bed.)		300							
Kitchen (Appliances)		1,200							
Sensible (Heat Gain & Heat Loss									
Sensible & Latent		1.30							
Duct Loss		1.10				1.10			
Capacity Multiplier		1.15							
Estimated Load (BTUH div. by 12,000		TONS				BTUH 3413			
Capacity Eqv. Selec.		TONS				KW			

Factors under D.T. 105 degrees based as follows:
Ceiling R-19/Walls R-11/Glass-single pain/Floor Slab, no insl.

Factors on this sheet have been selected from Manual J. Consult Manual J for types of construction not covered by these factors.

T.D. #860 /D.C.Howard

ADDITIONAL FACTORS TO APPLY IN BLANK COLUMNS V-S-61

SOURCE OF HEAT GAIN OR LOSS		COOLING		HEATING
	INSULATION	FACTORS	FACTORS	FACTORS
DESIGN TEMPERATURE		@100°	@105°	@15°
CEILINGS: UNDER UNCONDITIONED	NONE	11.5	11.6	36
SPACE OR VENTILATED	R11 - 3.0"	3.0	3.1	5.0
ATTIC	R13 - 3.6"	2.5	2.6	4.0
FOR COMBINATION	R19 - 6.0"	2.0	2.0	3.0
ROOF/CEILING ADD	R22 - 7.0"	1.7	1.8	2.5
0.3 to FACTOR LISTED	R26 - 8.0"	1.6	1.7	2.2
	R30 - 9.0"	1.4	1.5	1.9
	R33 -10.0"	1.3	1.4	1.7
	R44 -12.0"	1.0	1.1	1.3

		COOLING				HEATING	
GLASS: WINDOWS OR		100° D. TEMP.		105° D. TEMP.		15° D. TEMP.	
DOORS WITH	FACING	SINGLE	DOUBLE	SINGLE	DOUBLE	SINGLE	DOUBLE
DRAPES OR BLINDS	N	30	20	30	20	80	50
HALF DRAWN.	NE/NW	45	35	50	35	"	"
	E/W	60	50	65	50	"	"
	SE/SW	55	45	55	45	"	"
	S	35	30	40	30	"	"

		COOLING				HEATING	
GLASS: CLEAR STORY,		100° D. TEMP.		105° D. TEMP.		15° D. TEMP.	
NO SHADING.	FACING	SINGLE	DOUBLE	SINGLE	DOUBLE	SINGLE	DOUBLE
SKYLITES: FOR CLEAR GLASS	N	35	25	40	30	80	54
OR PLEXI., NO	NE/NW	70	55	70	55	"	"
SHADING, MULT.	E/W	95	75	95	80	"	"
FACTOR SHOWN	SE/SW	85	65	85	70	"	"
BY 1.50.	S	55	40	55	45	"	"

WALLS: WOOD FRAME,	R VALUE	COOLING		HEATING
WITH SHEETROCK,		100° D. TEMP.	105° D. TEMP.	15° D. TEMP.
½" SHEATHING,	0	9.0	9.0	15
BRICK OR WOOD	R11-3.0"	4.0	4.0	4.0
VENEER.	R13-3.6"	3.5	4.0	4.0
	R19-6.0"	3.0	3.3	2.4
WALLS: AS ABOVE WITH 1"				
POLYSTYRENE IN	R13+R5	3.3	3.6	3.0
LIEU GYP. SHEATH'G				

NOTE: INFILTRATION HAS BEEN INCLUDED IN HTM FACTORS.

CAPACITY MULTIPLIERS: 3° TEMP. SWING @100° = 1.10
101° = 1.11
105° = 1.15

SIZING, DESIGN AND RETROFIT

THERMAL LOAD ANALYSIS - SPACE HEATING AND SPACE COOLING

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

THERMAL LOAD ANALYSIS - SPACE HEATING AND SPACE COOLING

There are several reasons for analyzing the thermal loads. A quick look at the Btu/ft^2 of living space will provide a yardstick with which to measure the energy efficiency of the building. Also, one can readily see which zones within the building have high load factors. A closer examination may reveal that more extensive conservation measures would give a reasonable payback.

An example of zone loads - heating and cooling - is shown on the following page. One can readily see that the Recreation room and Work room have rather high heating loads. These two rooms account for 40% of the total heat loss. At this point we can examine the structure, floor plan and orientation to determine what, if anything, can be done to reduct the heat loss in these rooms. Obviously these rooms are going to be used quite extensively but any conservation measures that can be implemented here would certainly have a good payback, since they represent such a large percentage of the total load.

A closer look at the components within these spaces that contribute the need to the total room load reveals that 75% of the load in the Recreation room is from an above grade, uninsulated basement walls and casement windows. This load can certainly be reduced. The same can be done for the work room, and should be done before solarization of this house.

See heat loss calculations of following page.

V-S-66

1	Name of Room					Entire House			1 LIV			2 HALL'D' & DIN			3 LAUN			4 HALL'C' & KIT			5 BATH #1		
2	Running Ft Exposed Wall								21			12 & 13			11 & 7			11			9		
3	Room Dimensions, Ft								14 x 21			7x18 & 4x4			7x11			8x11 & 3x17			6x8 & 3x5		
4	Ceiling Ht, Ft	Directions Room Faces							8	W		8	N & W		8	N & E		8	E		8	E	
	Type of Exposure		Const No	HTM Htg	HTM Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg
5	Gross Exposed Walls and Partitions	a	8(d)						168			104			56			88			72		
		b	10(c)									96			88								
		c	11(a)																				
		d	11(f)																				
6	Windows and Glass Doors (Htg)	a	2(c)	70														11	800		8	600	
		b	4(b)	70					40	2800		20	1400										
		c	3(a)	205																			
7	Windows and Glass Doors (Clg)	North																					
		E & W or NE & NW																					
		South or SE & SW																					
8	Other Doors		7(b)	195											17	3300							
9	Net Exposed Walls and Partitions	a	8(d)	6					128	800		84	500		39	200		77	500		64	400	
		b	10(c)	10								96	1000		88	900							
		c	11(a)	38																			
		d	11(f)	5																			
10	Ceilings	a	12(c)	6					294	1800		142	900		77	500		139	800		63	400	
		b																					
11	Floors	a	17	3																			
		b																					
12	Ventilation		22	85		60	5100																
13	Sub Total Btuh Loss						5100			5400			3800			4900			2100			1400	
14	Duct Btuh Loss																						
15	Total Btuh Loss						69,400																
16	People @ 300 and Appliances 1200																						
17	Sensible Btuh Gain (Structure)																						
18	Duct Btuh Gain																						
19	Sum of Lines 17 and 18 (Clg)																						
20	Total Btuh Gain (Line 19 x 1.3)			1.3																			
21	Btuh for Air Quantities																						

1	6 BR #3			7 BR #2			8 BATH #2			9 BR #1			10 HALL'A' & 'B'			11 REC RM			12 WORK & UTIL		
2	10			24			5			29			7			82			74		
3	10x11 & 3x6			10x14 & 2x5			5x5			14x15, 2x5 & 3x6			4x14, 5x5, 4x20 4x4, 1x5 & 2x3			23x28 & 8x16			19x28 & 8x12		
4	8	E		8	S & E		8	S		8	S & W		8	W		3/5	E, S & W		3/5	W, N & E	
	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg	Area or Length	Btuh Htg	Btuh Clg
5	80			192			40			232			56								
																246			222		
																410			370		
6	22	1500		28	2000		8	600		28	2000										
																16	3300		4	800	
7																					
8													20	3900							
9	58	300		164	1000		32	200		204	1200		36	200							
																250	8700		218	8300	
																410	2100		370	1900	
10	128	800		150	900		25	200		238	1400		168	1000		28	200				
11																772	2300		628	1900	
12																					
13		2600			3900			1000			4600			5100			16,600			12,900	
14																					
15																					
16																					
17																					
18																					
19																					
20																					

One method of evaluating the effects of conservation measures is to do a short form calculation "as is" (without conservation measures) and one with conservation. This procedure requires only a few minutes and definitely gives the consumer the "bottom line" on energy cost comparisons.

Compare the heating and cooling loads and energy costs on the same house - with and without conservation measures - by examining the load forms below.

FORM 219R 5-76

EXAMPLE I

Texas Power & Light Company

HEATING AND COOLING ESTIMATE
RESIDENTIAL

Customer GOLD MEDALLION
Address PLAN # 1658 (HEAT PUMP)
Town IRVING District IRVING
Prepared by: M.T. Date 6-10-76

DESIGN CONDITIONS
COOLING D.B. °F.
Outside Temp. 100
Inside Temp. 75
Temp. Diff. 25

HEATING D.B. °F.
Inside Temp. 75
Outside Temp. 10
Temp. Diff. 65

INSULATION
Ceiling R-19
Wall R-11
Floor SLAB R-0
Window SINGLE

HOUSE FACES
☐ North
☐ East
☐ South
☒ West

OVERHANG OR SHADING
North 2'
East 2'
South 2'
West 2'

CONSTRUCTION	COOLING FACTOR							HEATING FACTOR	QUANTITY	COOLING BTUH	HEATING BTUH
1. GLASS—SOLAR GAIN Windows—Doors (Sq. Ft. (*Note A)	No Shading	Inside Shading & Overhang					Outside Shades				
		0'	1'	2'	3'	4'					
North	32	14	0	0	0	0	0		66	0	
Northeast	50	23	21	19	19	17	17				
East	50	23	20	19	17	14	13		18	342	
Southeast	80	36	26	19	8	0	21				
South	100	45	33	10	0	0	26		23	230	
Southwest	155	70	51	39	17	0	42				
West	190	86	82	80	67	57	57		108	8640*	
Northwest	128	58	54	51	51	49	37				
Horizontal	185										

	Uo	DESIGN D.B. TEMP. DIFFERENTIAL								QUANTITY	COOLING BTUH	HEATING BTUH
		20	22	25	30	65	60	55	50			
2 GLASS TRANSMISSION (Sq. Ft.)		Uo x TD				Uo x TD						
Standard—Single Glazing	1.13	22.6	24.9	28.2	33.9	73.4	67.8	62.2	56.5	215	6063	15781
Insulating—Double Glazing	.78	15.6	17.2	19.5	23.4	50.7	46.8	42.9	39.0			
Storm Window	.67	13.4	14.7	16.8	20.1	43.6	40.2	36.8	33.5			
Storm or Insulating Glass with Thermal Break	.56	11.2	12.3	14.0	16.8	36.4	33.6	30.8	28.0			
3 DOORS—(Sq. Ft.)		Uo x TD				Uo x TD						
Solid Wood or Hollow Core	.55	11.0	12.1	13.8	16.5	35.8	33.0	30.3	27.5	39	538	1396
Wood with Storm Door	.34	6.8	7.5	8.5	10.2	22.1	20.4	18.7	17.0			
Metal with 1½" Urethane	.11	2.2	2.4	2.8	3.3	7.2	6.7	6.1	5.6			
4a FLOOR: SLAB (Linear Ft. Exposed Edge)		Uo x zero				Uo x TD						
No Edge Insulation	.81					52.6	48.6	44.6	40.5	182		9573
R 4 Edge Insulation	.68					44.2	40.8	37.4	34.0			
R 7 Edge Insulation	.55					35.8	33.0	30.2	27.5			
4b FLOORS: ENCLOSED CRAWL SPACE (Sq. Ft.)		Uo x zero				Uo x (TD-20)						
No Insulation	.270					12.2	10.8	9.4	8.1			
R 7 Insulation	.093					4.2	3.7	3.2	2.8			
R 11 Insulation	.073					3.3	2.9	2.6	2.2			
R 13 Insulation	.060					2.7	2.4	2.1	1.8			
R 19 Insulation	.046					2.1	1.8	1.6	1.4			
R 22 Insulation	.039					1.8	1.6	1.4	1.2			
4c FLOORS: OPEN CRAWL SPACE (Sq. Ft.)		Uo x (TD-5)				Uo x TD						
No Insulation	.374	5.6	6.4	7.5	9.4	24.3	22.4	20.6	18.7			
R 7 Insulation	.103	1.5	1.8	2.1	2.6	6.7	6.2	5.7	5.2			
R 11 Insulation	.073	1.1	1.2	1.5	1.8	4.7	4.4	4.0	3.6			
R 13 Insulation	.064	1.0	1.1	1.3	1.6	4.2	3.8	3.5	3.2			
R 19 Insulation	.046	0.7	0.8	0.9	1.2	3.0	2.8	2.5	2.3			
R 22 Insulation	.041	0.6	0.7	0.8	1.0	2.7	2.5	2.3	2.0			
5 WALLS—(Sq. Ft.) NET		Uo x TD				Uo x TD						
No Insulation—Solid Masonry	.389	7.8	8.6	9.7	11.7	25.3	23.3	21.4	19.5			
No Insulation—Wood Siding	.320	6.4	7.0	8.0	9.6	20.8	19.2	17.6	16.0			
No Insulation—Brick Veneer	.240	4.8	5.3	6.0	7.2	15.6	14.4	13.2	12.0			
R 5 Insulation	.128	2.6	2.8	3.2	3.8	8.3	7.7	7.0	6.4			
R 7 Insulation	.109	2.2	2.4	2.7	3.3	7.1	6.5	6.0	5.5	1202	2284	5890
R 11 Insulation	.075	1.5	1.7	1.9	2.3	4.9	4.5	4.1	3.8			
R 13 Insulation	.065	1.3	1.4	1.6	2.0	4.2	3.9	3.6	3.3			
R 16 Insulation	.054	1.1	1.2	1.4	1.6	3.5	3.2	3.0	2.7			
R 19 Insulation	.047	0.9	1.0	1.2	1.4	3.1	2.8	2.6	2.4			
R 24 Insulation	.038	0.8	0.8	1.0	1.1	2.5	2.3	2.1	1.9			

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V-S-68

CONSTRUCTION	Uo	COOLING FACTOR DESIGN D.B. TEMP. DIFFERENTIAL 20	 22	 [✓ 25]	 30	HEATING FACTOR [✓ 65]	 60	 55	 50	QUANTITY	COOLING BTUH	HEATING BTUH
6a. CEILINGS—WITH ATTIC (Sq. Ft.) (*Note B)		Uo x (TD + 40)				Uo x TD						
No Insulation	[illegible]	35.9	37.1	38.9	41.9	38.9	35.9	32.9	29.9			
R 4 Insulation	.176	10.6	10.9	11.4	12.3	11.4	10.6	9.7	8.8			
R 7 Insulation	.114	6.8	7.1	7.4	8.0	7.4	6.8	6.3	5.7			
R 11 Insulation	.079	4.7	4.9	5.1	5.5	5.1	4.7	4.3	4.0			
R 19 Insulation	[illegible]	2.9	3.0	[✓ 3.1]	3.4	[✓ 3.1]	2.9	2.6	2.4	1658	5140	5140
R 22 Insulation	[illegible]	2.5	2.6	2.7	2.9	2.7	2.5	2.3	2.1			
R 26 Insulation	[illegible]	2.2	2.2	2.3	2.5	2.3	2.2	2.0	1.8			
R 30 Insulation	[illegible]	1.9	2.0	2.1	2.2	2.1	1.9	1.8	1.6			
R 33 Insulation	[illegible]	1.7	1.8	1.9	2.0	1.9	1.7	1.6	1.5			
R 38 Insulation	.025	1.5	1.6	1.6	1.8	1.6	1.5	1.4	1.2			
6b. CEILINGS—NO ATTIC (Sq. Ft.) (*Note B)		Uo x (TD + 45)				Uo x TD						
No Insulation	[illegible]	30.6	31.5	32.9	35.3	30.6	28.2	25.9	23.5			
R 4 Insulation	[illegible]	10.4	10.7	11.2	12.0	10.4	9.6	8.8	8.0			
R 5 Insulation	.120	8.5	8.7	9.1	9.8	8.5	7.8	7.2	6.5			
R 7 Insulation	[illegible]	7.1	7.3	7.6	8.2	7.1	6.5	6.0	5.5			
R 11 Insulation	.076	4.9	5.1	5.3	5.7	4.9	4.6	4.2	3.8			
R 19 Insulation	[illegible]	3.1	3.1	3.3	3.5	3.1	2.8	2.6	2.4			
R 26 Insulation	[illegible]	2.3	2.3	2.4	2.6	2.3	2.1	1.9	1.8			
R 30 Insulation	[illegible]	2.0	2.1	2.2	2.3	2.0	1.9	1.7	1.6			
7. SENSIBLE INFILTRATION VOL. METHOD (Cu. Ft.) (*Note C)		Uo x TD				Uo x TD						
0.50 Well Above Standard	[illegible]	0.18	0.20	0.22	0.27	0.58	0.54	0.50	0.45			
0.67 Above Standard	.012	0.24	0.26	0.30	0.36	0.78	0.72	0.66	0.60			
0.75 Standard	[illegible]	0.26	0.29	[✓ 0.32]	0.39	[✓ 0.84]	0.78	0.72	0.65	13264	4244	11142
1.00 Average	[illegible]	0.36	0.40	0.45	0.54	1.17	1.08	0.99	0.90			
1.50 Below Average	.027	0.54	0.59	0.68	0.81	1.76	1.62	1.48	1.35			
2.00 Well Below Average	[illegible]	0.72	0.79	0.90	1.08	2.34	2.16	1.98	1.80			
8. PEOPLE—SENSIBLE ONLY (Avg. No.)				[✓ 250]						4	1000	
9. **SUB TOTAL—SENSIBLE HEAT** (Total 1 through 8)											27909	48922
DUCT FACTOR (From Table below)											x 1.10	x 1.15
10. **TOTAL SENSIBLE—INCLUDING DUCT LOSS** (Multiply Line 9 Sub Total by Duct Factor)											30700	56260
11. **INFILTRATION—LATENT HEAT** (Cu. Ft.)				FACTOR								
0.50 Well Above Standard				0.332								
0.67 Above Standard				0.444								
0.75 Standard				[✓ 0.497]						13264	6592	
1.00 Average				0.633								
1.50 Below Average				0.994								
2.00 Well Below Average				1.333								
12. **PEOPLE—LATENT HEAT** (Avg. No.)				[✓ 200]						4	800	
13. **TOTAL LOAD BTUH** (Total 10, 11, 12)											38092	56260
14. **UNITARY COOLING EQUIPMENT REQUIREMENT** (BTUH); Line 10 x 1.25											38375	
15. **INSTALLED EQUIPMENT CAPACITY** (BTUH) Use Larger of Line 13 or Line 14											38375	56260
(*Note G.)											3.19 TONS	16.48 KW

COST OF OPERATION ESTIMATE FOR HEATING AND COOLING

COOLING 38.092 MBTUH (Line 13) ÷ 7 EER = 5.44 KW × 1700 Hours (Use 1700 Hrs. for 75° Maintained; Use 1400 Hrs. for 78° Maintained) (*Note D) = 9248 KWH/Season @ $ .03 /KWH = $ 277 /Season

HEATING 56.260 MBTUH (Line 13) × 287 KWH/MBTU (*Note E) ÷ 2.0 S.P.F. (Use 1.0 if Furnace; Use 2.0 if Heat Pump) (*Note F) = 8073 KWH/Season @ $ .02 /KWH = $ 161 /Season

Total $ 438 /Year

DUCTWORK LOCATION*	INSUL. THICKNESS	COOLING FACTOR	HEATING
Attic—vented	1"	1.15	1.25
—vented	2"	[✓ 1.10]	[✓ 1.15]
—unvented	1"	1.20	1.20
—unvented	2"	1.15	1.10
Crawl space—vented	2"	1.10	1.15
—unvented	1"	1.05	1.10
Within conditioned area	1"	1.00	1.00
within slab		1.20	1.25

*Note A 1. Use only the one largest Solar Gain
2. Multiply the Factors by .90 for Plate Glass, .85 for double glass
3. No Solar Gain is used if overhang exceeds 4', or if glass is shaded by permanent structure

*Note B For light colored roof, Multiply COOLING Factor by .75

*Note C See infiltration Construction Definitions (See pad back)

*Note D Full load equivalent run hours

*Note E Annual Heating Consumption Factor from TP&L Service area Weather Data (See pad back)

*Note F S.P.F. is Seasonal Performance Factor of Heating Equipment

*Note G 12,000 BTU/TON, 3413 BTU/KW

FORM 719R 5 76

EXAMPLE II

Texas Power & Light Company

HEATING AND COOLING ESTIMATE
RESIDENTIAL

Customer ENERGY EFFICIENT
Address PLAN #1658 (RESISTANCE)
Town IRVING District IRVING
Prepared by: M.T. Date 6-10-76

DESIGN CONDITIONS
COOLING D.B. °F.
Outside Temp. 100
Inside Temp. 75
Temp. Diff. 25

HEATING D.B. °F.
Inside Temp. 75
Outside Temp. 10
Temp. Diff. 65

INSULATION
Ceiling R-26
Wall R-16
Floor SLAB R-0
Window STORM W/T.B

HOUSE FACES
☐ North
☐ East
☐ South
☒ West

OVERHANG OR SHADING
North 2'
East 2'
South 2'
West 2'

CONSTRUCTION	COOLING FACTOR							HEATING FACTOR	QUANTITY	COOLING BTUH	HEATING BTUH
1 **GLASS—SOLAR GAIN Windows—Doors** (Sq. Ft. (*Note A)	No Shading	Inside Shading & Overhang 0'	1'	2'	3'	4'	Outside Shades				
North	32	14	0	0	0	0	0		66	0	
Northeast	50	23	21	19	19	17	17				
East	50	23	20	19	17	14	13		18	291	
Southeast	80	36	26	19	8	0	21				
South	100	45	33	10	0	0	26		23	196	
Southwest	155	70	51	39	17	0	42				
West	190	86	82	80	67	57	57		108	7344*	
Northwest	128	58	54	51	51	49	37				
Horizontal	185										

	Uo	DESIGN D.B. TEMP. DIFFERENTIAL 20	22	25	30	65	60	55	50	QUANTITY	COOLING BTUH	HEATING BTUH
2 **GLASS TRANSMISSION** (Sq. Ft.)		Uo x TD				Uo x TD						
Standard—Single Glazing	1.13	22.6	24.9	28.2	33.9	73.4	67.8	62.2	56.5			
Insulating—Double Glazing	.78	15.6	17.2	19.5	23.4	50.7	46.8	42.9	39.0			
Storm Window	.67	13.4	14.7	16.8	20.1	43.6	40.2	36.8	33.5			
Storm or Insulating Glass with Thermal Break	.56	11.2	12.3	14.0	16.8	36.4	33.6	30.8	28.0	215	3010	7826
3 **DOORS**—(Sq. Ft.)		Uo x TD				Uo x TD						
Solid Wood or Hollow Core	.55	11.0	12.1	13.8	16.5	35.8	33.0	30.3	27.5			
Wood with Storm Door	.34	6.8	7.5	8.5	10.2	22.1	20.4	18.7	17.0	39	538	1396
Metal with 1½" Urethane	.11	2.2	2.4	2.8	3.3	7.2	6.7	6.1	5.6			
4a **FLOOR: SLAB** (Linear Ft. Exposed Edge)		Uo x zero				Uo x TD						
No Edge Insulation	.81					52.6	48.6	44.6	40.5			
R 4 Edge Insulation	.68					44.2	40.8	37.4	34.0	182		9573
R 7 Edge Insulation	.55					35.8	33.0	30.2	27.5			
4b **FLOORS: ENCLOSED CRAWL SPACE** (Sq. Ft.)		Uo x zero				Uo x (TD-20)						
No Insulation	.270					12.2	10.8	9.4	8.1			
R 7 Insulation	.093					4.2	3.7	3.2	2.8			
R 11 Insulation	.073					3.3	2.9	2.6	2.2			
R 13 Insulation	.060					2.7	2.4	2.1	1.8			
R 19 Insulation	.046					2.1	1.8	1.6	1.4			
R 22 Insulation	.039					1.8	1.6	1.4	1.2			
4c **FLOORS: OPEN CRAWL SPACE** (Sq. Ft.)		Uo x (TD-5)				Uo x TD						
No Insulation	.374	5.6	6.4	7.5	9.4	24.3	22.4	20.6	18.7			
R 7 Insulation	.103	1.5	1.8	2.1	2.6	6.7	6.2	5.7	5.2			
R 11 Insulation	.073	1.1	1.2	1.5	1.8	4.7	4.4	4.0	3.6			
R 13 Insulation	.064	1.0	1.1	1.3	1.6	4.2	3.8	3.5	3.2			
R 19 Insulation	.046	0.7	0.8	0.9	1.2	3.0	2.8	2.5	2.3			
R 22 Insulation	.041	0.6	0.7	0.8	1.0	2.7	2.5	2.3	2.0			
5 **WALLS**—(Sq. Ft.) NET		Uo x TD				Uo x TD						
No Insulation—Solid Masonry	.389	7.8	8.6	9.7	11.7	25.3	23.3	21.4	19.5			
No Insulation—Wood Siding	.320	6.4	7.0	8.0	9.6	20.8	19.2	17.6	16.0			
No Insulation—Brick Veneer	.240	4.8	5.3	6.0	7.2	15.6	14.4	13.2	12.0			
R 5 Insulation	.128	2.6	2.8	3.2	3.8	8.3	7.7	7.0	6.4			
R 7 Insulation	.109	2.2	2.4	2.7	3.3	7.1	6.5	6.0	5.5			
R 11 Insulation	.075	1.5	1.7	1.9	2.3	4.9	4.5	4.1	3.8			
R 13 Insulation	.065	1.3	1.4	1.6	2.0	4.2	3.9	3.6	3.3			
R 16 Insulation	.054	1.1	1.2	1.4	1.6	3.5	3.2	3.0	2.7	1202	1683	4207
R 19 Insulation	.047	0.9	1.0	1.2	1.4	3.1	2.8	2.6	2.4			
R 24 Insulation	.038	0.8	0.8	1.0	1.1	2.5	2.3	2.1	1.9			

V-S-70

CONSTRUCTION	Uo	COOLING FACTOR				HEATING FACTOR				QUANTITY	COOLING BTUH	HEATING BTUH
		DESIGN D.B. TEMP. DIFFERENTIAL										
		20	22	[✓ 25]	30	[✓ 65]	60	55	50			
6a. **CEILINGS—WITH ATTIC** (Sq. Ft.) (*Note B)		Uo x (TD + 40)										
No Insulation		35.9	37.1	38.9	41.9	38.9	35.9	32.9	29.9			
R-4 Insulation		10.6	10.9	11.4	12.3	11.4	10.6	9.7	8.8			
R-7 Insulation		6.8	7.1	7.4	8.0	7.4	6.8	6.3	5.7			
R-11 Insulation		4.7	4.9	5.1	5.5	5.1	4.7	4.3	4.0			
R-19 Insulation		2.9	3.0	3.1	3.4	3.1	2.9	2.6	2.4			
R-22 Insulation		2.5	2.6	2.7	2.9	2.7	2.5	2.3	2.1			
R-26 Insulation		2.2	2.2	[✓ 2.3]	2.5	[✓ 2.3]	2.2	2.0	1.8	1658	3813	3813
R-30 Insulation		1.9	2.0	2.1	2.2	2.1	1.9	1.8	1.6			
R-33 Insulation		1.7	1.8	1.9	2.0	1.9	1.7	1.6	1.5			
R-38 Insulation		1.5	1.6	1.6	1.8	1.6	1.5	1.4	1.2			
6b. **CEILINGS—NO ATTIC** (Sq. Ft.) (*Note B)		Uo x (TD + 45)				Uo x TD						
No Insulation		30.6	31.5	32.9	35.3	30.6	28.2	25.9	23.5			
R-4 Insulation		10.4	10.7	11.2	12.0	10.4	9.6	8.8	8.0			
R-5 Insulation		8.5	8.7	9.1	9.8	8.5	7.8	7.2	6.5			
R-7 Insulation		7.1	7.3	7.6	8.2	7.1	6.5	6.0	5.5			
R-11 Insulation		4.9	5.1	5.3	5.7	4.9	4.6	4.2	3.8			
R-19 Insulation		3.1	3.1	3.3	3.5	3.1	2.8	2.6	2.4			
R-26 Insulation		2.3	2.3	2.4	2.6	2.3	2.1	1.9	1.8			
R-30 Insulation		2.0	2.1	2.2	2.3	2.0	1.9	1.7	1.6			
7. **SENSIBLE INFILTRATION VOL. METHOD** (Cu. Ft.) (*Note C)		Uo x TD				Uo x TD						
0.50 Well Above Standard		0.18	0.20	[✓ 0.22]	0.27	[✓ 0.58]	0.54	0.50	0.45	13264	2918	7693
0.67 Above Standard		0.24	0.26	0.30	0.36	0.78	0.72	0.66	0.60			
0.75 Standard		0.26	0.29	0.32	0.39	0.84	0.78	0.72	0.65			
1.00 Average		0.36	0.40	0.45	0.54	1.17	1.08	0.99	0.90			
1.50 Below Average		0.54	0.59	0.68	0.81	1.76	1.62	1.48	1.35			
2.00 Well Below Average		0.72	0.79	0.90	1.08	2.34	2.16	1.98	1.80			
8. **PEOPLE—SENSIBLE ONLY** (Avg. No.)				[✓ 250]						4	1000	
9. **SUB TOTAL—SENSIBLE HEAT** (Total 1 through 8)											20306	34508
DUCT FACTOR (From Table below)											x 1.10	x 1.15
10. **TOTAL SENSIBLE—INCLUDING DUCT LOSS** (Multiply Line 9 Sub Total by Duct Factor)											22337	39684
11. **INFILTRATION—LATENT HEAT** (Cu. Ft.)		FACTOR										
0.50 Well Above Standard		[✓ 0.332]								13264	4404	
0.67 Above Standard		0.444										
0.75 Standard		0.497										
1.00 Average		0.633										
1.50 Below Average		0.994										
2.00 Well Below Average		1.333										
12. **PEOPLE—LATENT HEAT** (Avg. No.)		[✓ 200]								4	800	
13. **TOTAL LOAD BTUH** (Total 10, 11, 12)											27541	39684
14. **UNITARY COOLING EQUIPMENT REQUIREMENT** (BTUH): Line 10 x 1.25											27921	
15. **INSTALLED EQUIPMENT CAPACITY** (BTUH) Use Larger of Line 13 or Line 14											27921	39684
(*Note G)											2.33	11.63
											TONS	KW

COST OF OPERATION ESTIMATE FOR HEATING AND COOLING

COOLING 27.541 MBTUH (Line 13) ÷ 8.0 EER = 3.44 KW × 1700 Hours (Use 1700 Hrs. for 75° Maintained; Use 1400 Hrs. for 78° Maintained; (*Note D)) = 5852 KWH/Season @ $.03 /KWH = $ 175 /Season

HEATING 39.684 MBTUH (Line 13) × 287 KWH/MBTU (*Note E) ÷ 1.0 S.P.F. (Use 1.0 if Furnace; Use 2.0 if Heat Pump; (*Note F)) = 11389 KWH/Season @ $.02 /KWH = $ 228 /Season

Total $ 403 /Year

DUCTWORK LOCATION*	INSUL. THICKNESS	COOLING FACTOR	HEATING
Attic—vented	1"	1.15	1.25
—vented	2"	[✓ 1.10]	[✓ 1.15]
—unvented	1"	1.20	1.20
—unvented	2"	1.15	1.10
Crawl space—vented	2"	1.10	1.15
—unvented	1"	1.05	1.10
Within conditioned area	1"	1.00	1.00
within slab		1.20	1.25

*Note A 1. Use only the one largest Solar Gain
2. Multiply the Factors by .90 for Plate Glass, .85 for double glass
3. No Solar Gain is used if overhang exceeds 4', or if glass is shaded by permanent structure
*Note B For light colored roof, Multiply COOLING Factor by .75
*Note C See infiltration Construction Definitions (See pad back)
*Note D Full load equivalent run hours
*Note E Annual Heating Consumption Factor from TP&L Service Area Weather Data (See pad back)
*Note F S.P.F. is Seasonal Performance Factor of Heating Equipment
*Note G 12,000 BTU/TON; 3413 BTU/KW

SIZING, DESIGN AND RETROFIT

THERMAL LOAD ANALYSIS - SERVICE WATER

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

THERMAL LOAD ANALYSIS - SERVICE WATER

The oldest and simplest domestic use of solar energy is for heating water. Solar hot water heaters were used in the United States at least 75 years ago, first in southern California and later in southern Florida. Although the use of solar water heaters in these regions declined during the last 40 years, use in Australia, Israel, and Japan has risen rapidly, particularly in the last 15 years.

In its simplest form, a solar water heater comprises a flat-plate water heating collector and an insulated storage tank positioned at a higher level than the collector. These components, connected to the cold water main and the hot water service piping in the dwelling, provide most of the hot water requirements in a sunny climate. Nearly all of the solar hot water systems used in the United States have been of this type.

The space heating load for a building is generally the major load required. While space heating is the major part of the total load, the domestic hot water heating load must also be considered. This domestic hot water load is difficult to estimate with any degree of accuracy, since the load varies with the personal living habits of occupants, as well as usage of automatic cycle washers, dishwashers, available supply water temperature, etc.

Water consumption studies conducted during the time period of the early 1960's illustrated that the average American family used about 50 gallons of hot water per day without the automatic cycle washer. When the automatic cycle washer was used, the average increased to about 75 gallons per day. The study also concluded that the average family used about 20 gallons of hot water per day per family member.

Other hot water usage tables are available, as in the ASHRAE Guide. These tables are generally directed at generous sizing of hot water heating equipment so as to meet absolute maximum possible hot water draw rates; therefore the tables may overstate the hot water need for the typical residence.

While reference must be made to specific family habits concerning water usage, residential solar system hot water draw rate is generally based on the following:

1. 50 gallons hot water used per day; no automatic cycle washer.
2. 75 gallons hot water used per day with automatic cycle washer.

OR

3. 20 gallons hot water used per day per family member. This is the estimate usually used in sizing DHW systems.

The daily hot water heating load will be dependent on the daily usage (draw rate) of hot water, the inlet cold water temperature (street or well) and the final hot water supply temperature. The relationship between these variables and hot water heat load can be expressed as follows:

$$\frac{\text{BTU}}{\text{Day}} = \frac{\text{Gal. Used}}{\text{Day}} \times \frac{8.33\ \text{lb.}}{\text{Gal.}} \times (140^{\circ\,*} - \text{Inlet Temperature})$$

* 140° F is the maximum recommended hot water temperature for residential applications.

For 100 Gal./Day usage rate and for 40° inlet temperature, the heat load will be:

$$\frac{\text{BTU}}{\text{Day}} = \frac{100\ \text{Gal.}}{\text{Day}} \times 8.33\ (140 - 40) = 83{,}000\ \frac{\text{BTU}}{\text{Day}}$$

It should be noted that cold inlet (street or well) water is commonly assumed to have a temperature of 40° F. In some areas of the country, this temperature will be higher and/or may be subject to seasonal change. The designer may adjust for changed inlet temperatures since this forecasts a changed hot water heating load. Month by month records of street main inlet water temperatures are generally available from water works and/or may be measured as from a well.

While an increase in cold inlet water temperature will reduce total heating need for domestic hot water, a decrease in the set tap water temperature will not necessarily decrease the heat load. The 140° tap water temperature from the water heater should be considered as fixed because the estimated volume usage for the building is based on 140° and on the mixture of 140° water with cold water supply to provide a usable temperature to the shower, wash basins, laundry tubs and the automatic washers. A heater set to provide 120° supply of water will, in other words, simply provide an increased volume of 120° relative to 140° to provide the same mixed temperature at the showers, etc.

The anticipated hot water heating load should be determined as follows. An example for 100 Gal. per day hot water consumption is shown below. It should be noted that inlet city water temperature variation has been used in this calculation.

Determine Monthly and Daily Domestic Hot Water Heat Load

				J	F	M	A	M	J	J	A	S	O	N	D
(12)	Gal. Hot Water Per Day	See Ref. for Judgement	$\frac{\text{Gal.}}{\text{Day}}$	100	→										
(13)	Base Delivered H. W. Temp.	140°; See Discussion	$^{\circ}$F	140	140	140	140	140	140	140	140	140	140	140	140
(14)	City or Well Temp.; $^{\circ}$F	City Records	$^{\circ}$F	35	35	37	42	52	56	58	61	66	59	51	39
(15)	Temp. Diff. (ΔT)	(13) - (14)	$^{\circ}$F	105	105	103	98	88	84	82	79	74	81	89	101
(16)	H. W. Heating Load/Day	$\frac{(12)\times(15)\times 8.33}{1000}$	1000 BTU/Day	88	88	86	82	73	70	68	66	62	67	74	84
(17)	H. W. Heating Load/Month	$\frac{(8)\times(16)}{1000}$	Million BTU/Mo.	2.7	2.5	2.7	2.5	2.3	2.1	2.1	2	1.8	2.1	2.2	2.6

TYPES AND CHARACTERISTICS OF SOLAR HOT WATER HEATERS

Most of the solar water heaters that have been experimentally and commercially used can be placed in two main groups:

1. Circulating types, involving the supply of solar heat to a fluid circulating through a collector and storage of hot water in a separate tank.

2. Non-circulating types, involving the use of water containers that serve both as solar collector and storage.

The circulating group may be divided into the following types and sub-types:

1. Direct heating, single-fluid types in which the water is heated directly in the collector, by:
   a. Thermosiphon circulation between collector and storage
   b. Pumped circulation between collector and storage.

2. Indirect heating, dual-fluid types in which a non-freezing medium is circulated through the collector for subsequent heat exchange with water, when:
   a. Heat transfer medium is a non-freezing liquid
   b. Heat transfer medium is air.

DIRECT HEATING, THERMOSIPHON CIRCULATING TYPE

The most common type of solar water heater, used almost exclusively in non-freezing climates, is shown in Figure 4-1. The collector, usually single glazed, may vary in size from about 30 square feet to 80 square feet, whereas the insulated storage tank is commonly in the range of 40 to 80 gallons capacity.

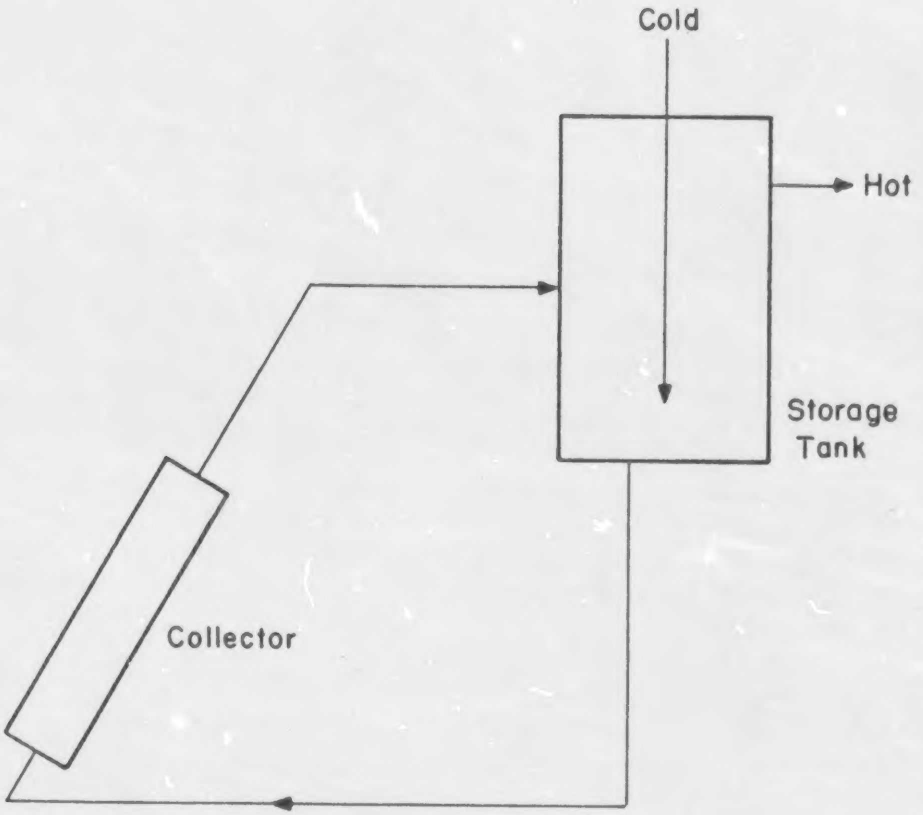


Figure 4-1. Direct Heating Thermosiphon Circulation Type of Solar Water Heater

The hot water requirements of a family of four persons can usually be met by a system in the middle of this size range, in a sunny climate. Operation at supply line pressure can be provided if the system is so designed. With a float valve in the storage tank or in an elevated head tank, unpressurized operation can be utilized, if the system is not designed for pressure. In the latter case, gravity flow from the hot water tank to hot water faucets would have to be accepted, or an automatic pump would have to be provided in the hot water line to supply pressure service. Plumbing systems and fixtures in the United States normally require the pressurized system.

Location of the tank higher than the top of the collector permits circulation of water from the bottom of the tank through the collector and back to the top of the tank. The density difference between cold and hot water produces the circulating flow. Temperature stratification in the storage tank permits operation of the collector under most favorable conditions, water at the lowest available temperature being supplied to the collector and the highest available temperature being provided to service. Circulation occurs only when solar energy is being received, to the system is self-controlling. The higher the radiation level, the greater the heating and the more rapid the circulating rate will be. In a typical collector under a full sun, a temperature rise of 15° F to 20° F is commonly realized in a single pass through the collector.

To prevent reserve circulation and cooling of stored water when no solar energy is being received, the bottom of the tank should be located at least 12 inches above the top header of the collector. If the collector is on a house roof, the tank may also be on the roof or in the attic space beneath a sloping roof. The roof (or attic) must be able to support this additional load at 8.33 lbs./gal. plus tank weight.

Although seldom used in cold climates, the thermosiphon type of solar water heater (storage tank above collector) can be protected from freezing by draining the collector. To avoid draining the storage tank also, thermostatically actuated valves in the lines between collector and storage tank must close when freezing threatens; a collector drain valve must open, and a collector vent valve must also open. The collector will then drain, and air will enter the collector tubes. Water in the storage tank, either inside the heated space or sufficiently well insulated to avoid freezing, does not enter the collector during the period when sub-freezing temperatures threaten. Resumption of operation requires closure of the drain and vent valves and opening of the valves in the circulating line. The possibility of control failure or valve malfunction makes this complex system unattractive in freezing climates.

DIRECT HEATING, PUMP CIRCULATION TYPES

If placement of the storage tank above the collector is inconvenient or impossible, the tank may be located below the collector and a small pump used for circulating water between collector and storage tank. This arrangement if usually more practical than the thermosiphon type in the United States, because the collector would often be located on the roof with a storage tank in the basement. Instead of thermosiphon circulation when the sun shines, a temperature sensor actuates a small pump, which circulates water through the collector-storage loop. A schematic arrangement is shown in Figure 4-2. To obtain maximum utilization of solar energy, control is based on the difference in water temperature at collector outlet and bottom of storage tank. Whenever this difference exceeds a preset number of degrees, say 10° F, the pump motor is actuated. The sensor at the collector outlet must be located close enough to the collector so that it is affected by collector temperature, even when the pump is not running. Similarly, the sensor

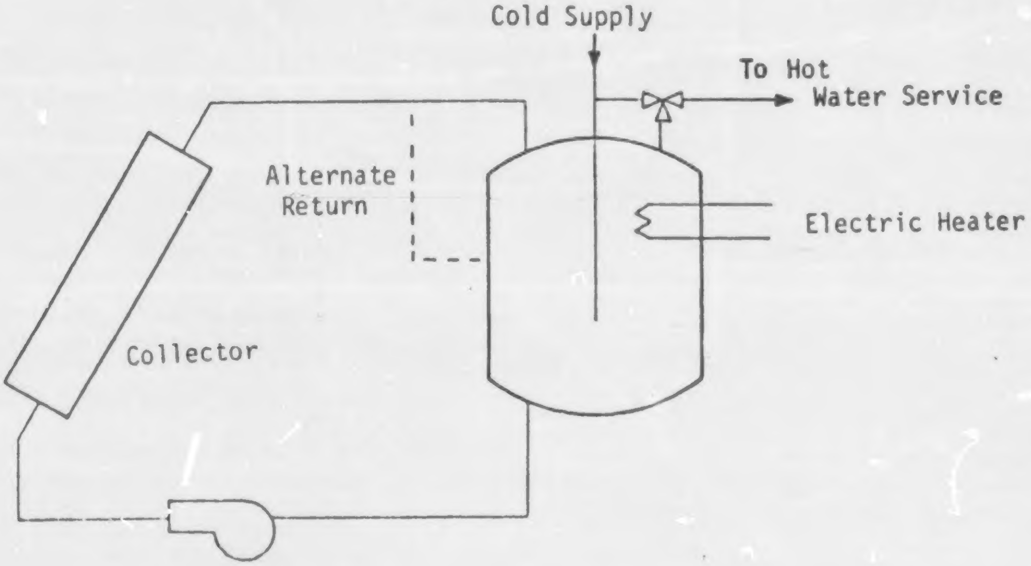


Figure 4-2. Direct Heating, Pump Circulation Type of Solar Water Heater

in the storage tank should be located in or near the bottom outlet from which the collector is supplied. When the temperature difference falls below the preset value, the pump is shut off and circulation ceases. To prevent reverse thermosiphon circulation and consequent water cooling when no solar energy is being received, a check valve should be located in the circulation line.

If hot water use is not sufficient to maintain storage tank temperature at normal levels (as during several days of non-use), boiling may occur in the collector. If a check valve or pressure-reducing valve prohibits back flow from the storage tank into the main, a relief valve must be provided in the collector-storage loop. The relief valve will permit the escape of steam and prevent damage to the system.

DIRECT HEATING, PUMP CIRCULATION, DRAINABLE TYPES

If the solar water heater described above is used in a cold climate, it may be protected from freeze damage by draining the collector when sub-freezing temperatures are encountered. Several methods can be used. Their common requirement, however, is reliability, even when electric power may not be available. One method is shown in Figure 4-3.

Drainage of the collector in freezing weather can be accomplished by automatic valves which provide water outflow to a drain (sewer) and the inflow of air to the collector. The control system can be arranged so that whenever the circulating pump is not in operation, these two valves are open. To assure maximum reliability, the valves should be mechanically driven to the drain position (by springs or other means), rather than electrically, so that in the event of a power failure, the collector can automatically drain.

The drainage system shown in Figure 4-3 is actuated by the temperature sensor, T_c, in the collector. When the sensor indicates a possibility of freezing, it can open the drainage and vent valves, thereby providing protection.

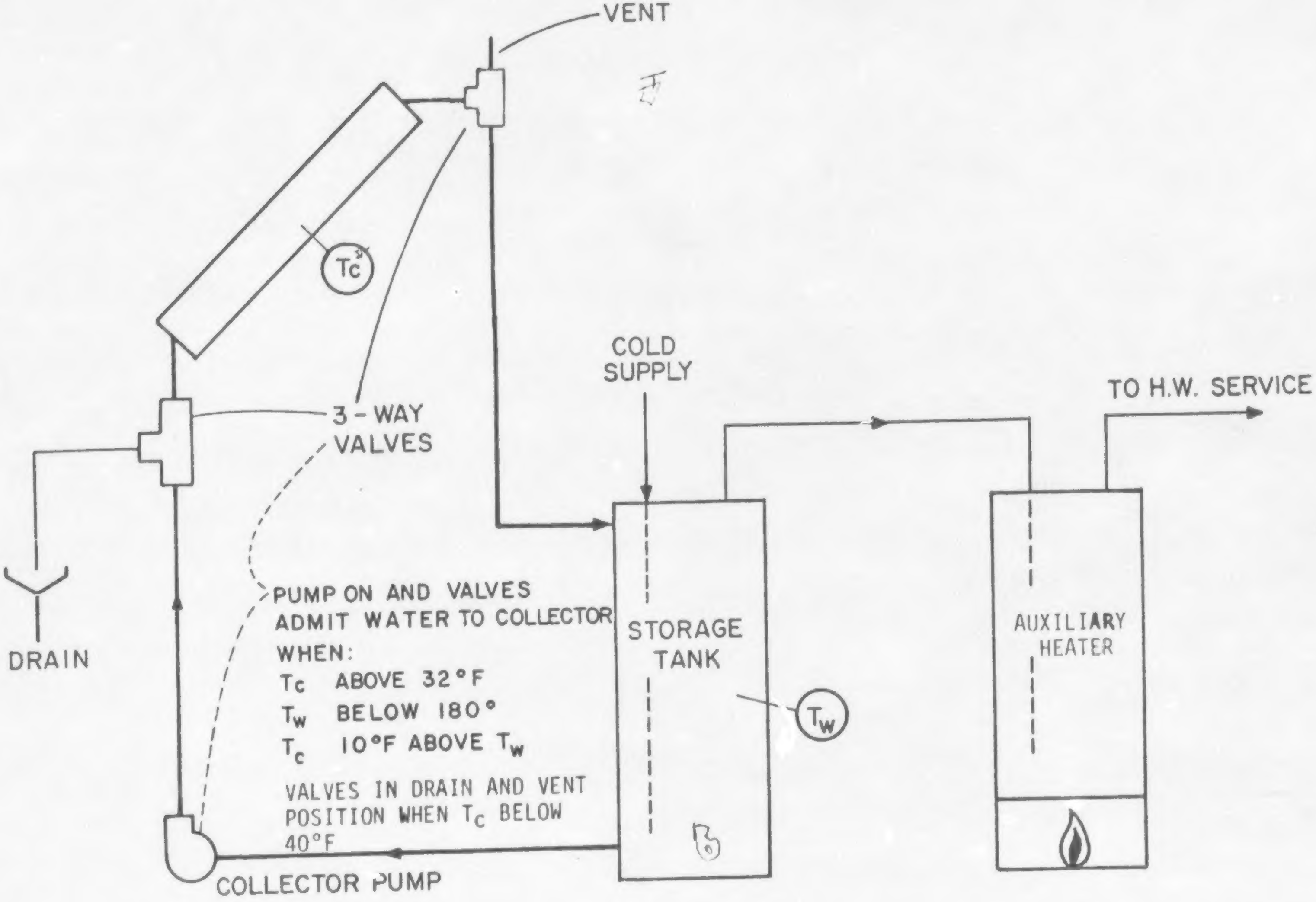

VENT
3 - WAY VALVES
T_c
COLD SUPPLY
TO H.W. SERVICE
DRAIN
PUMP ON AND VALVES ADMIT WATER TO COLLECTOR WHEN:
T_c ABOVE 32°F
T_w BELOW 180°
T_c 10°F ABOVE T_w
VALVES IN DRAIN AND VENT POSITION WHEN T_C BELOW 40°F
STORAGE TANK
T_w
AUXILIARY HEATER
COLLECTOR PUMP

Figure 4-3. Solar Water Heater with Freeze Protection by Automatic Collector Drainage.

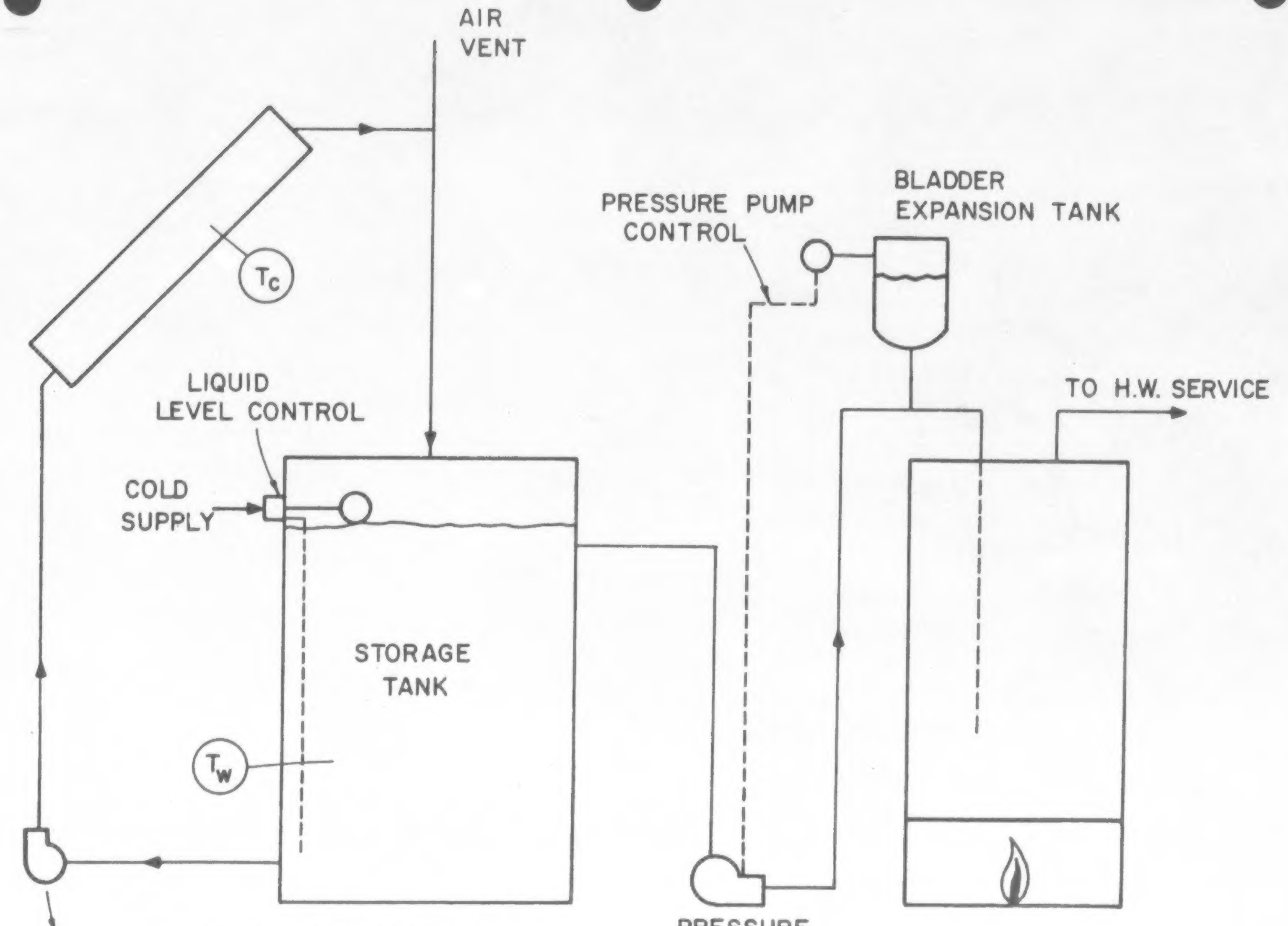
The temperature sensor can be of the vapor pressure type, with capillary tube connections to mechanical valve actuators, or of the electrical type where the valves are helped open by electrical means, automatically closing either when electrical failure occurs, or at low temperatures.

Another possibility for drainage of the collector is based on use of a non-pressurized collector and storage assembly as shown in Figure 4-4. A float valve in the storage tank controls the admission of cold water to the tank, and a pump in the hot water distribution system can furnish the necessary service pressure. With this design, the solar collector drains into the storage tank whenever the pump is not operating, as air enters the collector through a vent.

Start-up of any of the vented collector systems must permit the displacement of air from the collector. In either the line-pressure system or the unpressurized system, the entry of water into the collector (from the shut-off valve or pump) forces air from the collector tubes as long as the vent remains open. The vent valve design can be of a type which automatically passes air, but shuts off when water reaches it.

CIRCULATING TYPE, INDIRECT HEATING

As can be inferred from the above discussions of needs and means for collector drainage in freezing climates, costs and hazards are involved with those systems. The drainage requirement can be eliminated by the use of a non-freezing heat transfer medium in the solar collector, and a heat exchanger (inside the building) for transfer of heat from the solar heat collecting medium to the service water. The collector need never be drained, and there is no risk of freezing and damage. Corrosion rate in the wet collector tubes is also decreased when intermittent admission of oxygen is not required.


AIR VENT
PRESSURE PUMP CONTROL
BLADDER EXPANSION TANK
T_C
LIQUID LEVEL CONTROL
COLD SUPPLY
TO H.W. SERVICE
STORAGE TANK
T_W
PRESSURE PUMP

ON WHEN T_C IS AT LEAST 10°F ABOVE T_W
OFF WHEN T_W IS ABOVE 180° OR WHEN T_C
IS NOT AT LEAST 10°F ABOVE T_W

Figure 4-4. Unpressurized Vented Solar Water Heater System

Liquid Transfer Media

Figure 4-5 illustrates a method for solar water heating with a liquid heat transfer medium in the solar collector. The most commonly used liquid is a solution of ethylene glycol (which is common automobile radiator antifreeze) in water. A pump circulates this unpressurized solution, as in the direct water heating system, and delivers the liquid to and through a liquid-to-liquid heat exchanger. Simultaneously, another pump circulates domestic water from the storage tank through the exchanger, back to storage. The control system is essentially the same as that in the design employing water in the collector directly. If the heat exchanger is located below the bottom of the storage tank, and if the pipe sizes and heat exchanger design are adequate, thermosiphon circulation of water through the heat exchanger can be used. A small expansion tank needs to be provided in the collector loop, preferably near the high point of the system, with a vent to the atmosphere.

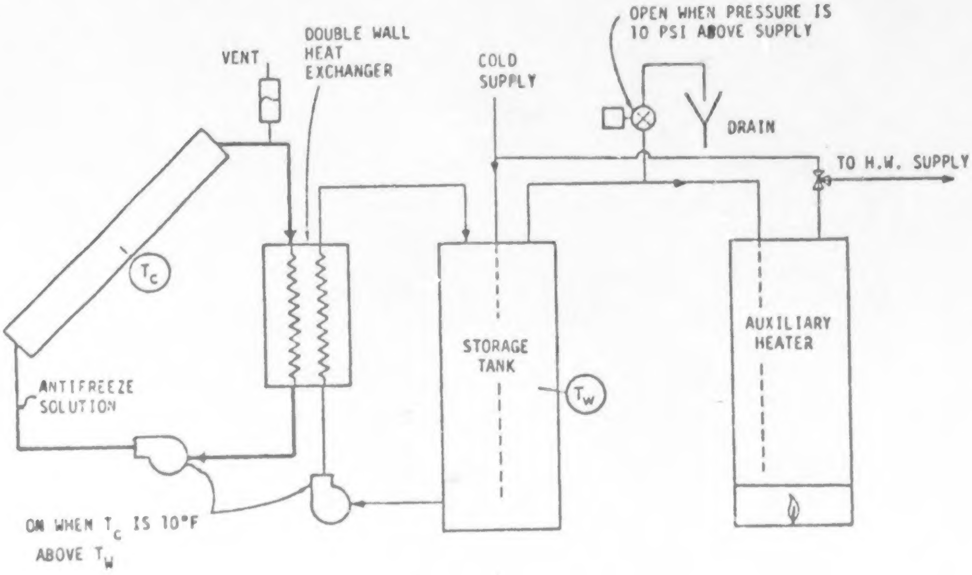

OPEN WHEN PRESSURE IS 10 PSI ABOVE SUPPLY
VENT
DOUBLE WALL HEAT EXCHANGER
COLD SUPPLY
DRAIN
TO H.W. SUPPLY
T_c
STORAGE TANK
AUXILIARY HEATER
T_w
ANTIFREEZE SOLUTION
ON WHEN T_c IS 10°F ABOVE T_w

Figure 4-5. Dual Liquid Solar Hot Water Heater

Although aqueous solutions of ethylene glycol and propylene glycol appear to be most practical for solar energy collection, organic liquids such as Dowtherm J and Therminol 55 may be employed. Price and viscosity are drawbacks, but chemical stability and assurance against boiling are advantages over the antifreeze mixtures.

Solar Collection in Heated Air

In a manner similar to that described immediately above, solar energy can be employed in an air heating collector with subsequent transfer to domestic water in an air-to-water exchanger. Figure 4-6 illustrates a method for employing this concept. A solar air heater is supplied with air from a blower, the air is heated by passage through the collector, the the hot air is then cooled in the heat exchanger through which domestic water from a storage tank is either being pumped or is circulating by thermosiphon action. Air from the heat exchanger is recirculated to the collector. Differential temperature control (between collector and storage) is employed as in the other systems described. Advantages of the air heat transfer medium are the absence of

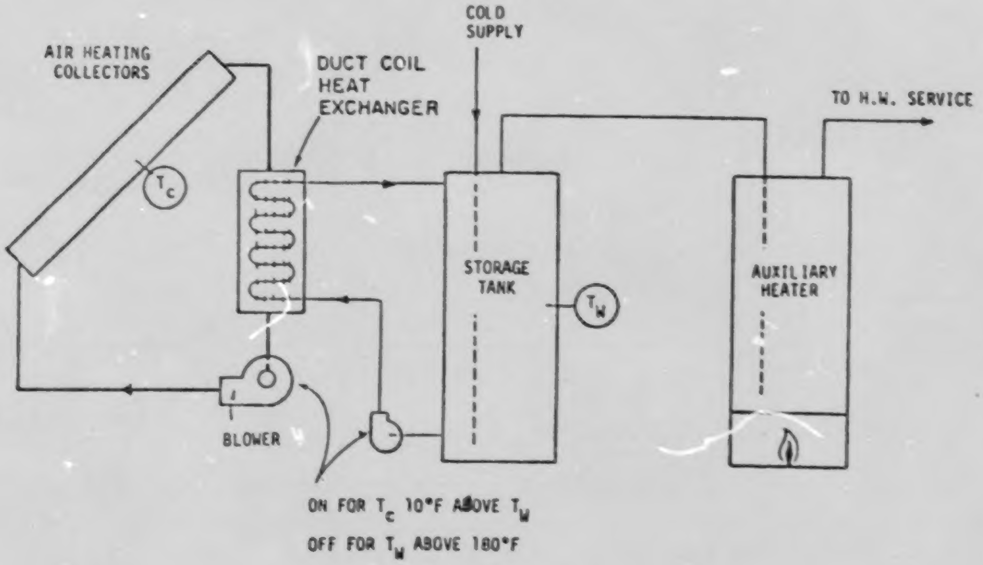

AIR HEATING COLLECTORS
T_c
DUCT COIL HEAT EXCHANGER
COLD SUPPLY
TO H.W. SERVICE
STORAGE TANK
T_W
AUXILIARY HEATER
BLOWER
ON FOR T_c 10°F ABOVE T_W
OFF FOR T_W ABOVE 180°F

Figure 4-6. Solar Hot Water Heater with Air Collectors

in the collector loop, freedom from liquid leakage, and freedom from boiling and loss of collector fluid. Disadvantages are the larger conduit between collector and heat exchanger, higher power consumption for circulation, and slightly larger collector surface requirements.

NON-CIRCULATING TYPE

Although probably of little potential interest in the United States, a type of solar water heater extensively used in Japan involves heat collection and water storage in the same unit. The most common type comprises a set of black plastic tubes about six inches in diameter and several feet long in a glass-covered box. Usually mounted in a tilted position, the tubes are filled each morning with water in which solar heat is collected throughout the day. The filling can be accomplished by a float-controlled valve and a small supply tank. Late in the day, heated water can be drained from the tubes for household use. In typical Japanese installations, non-pressurized hot water service is thus provided. Heat loss from the system is sufficiently high at night that hot water is usually not available until several hours after sunrise.

AUXILIARY HEAT

A dependable supply of hot water required the availability of auxiliary heat for supplementing the solar source. The numerous methods of providing auxiliary heat vary in cost and effectiveness. A general principle for maximizing solar supply and minimizing auxiliary use is the avoidance of direct or indirect auxiliary heat input to the fluid entering the solar collector. If auxiliary heat is added to the solar hot water storage tank, so that the temperature of the liquid supplied to the collector is increased above that which only the solar system would provide, efficiency is reduced because of higher heat losses from the collector. Thus, auxiliary heat should be added at a point beyond (downstream from) the solar collector-storage system. Figures 4-3 and 4-4 show a conventional gas-fired hot water heater being supplied with hot water from the solar tank (whenever a hot water tap is opened). Any deficiency in temperature is made up by fuel in the thermostatted conventional heater.

Alternatively, a "fast response", in-line heater can be employed. It is evident that auxiliary heat supply in these designs cannot adversely affect the operation of the solar system.

Another way in which auxiliary heat can be used without reducing solar collection efficiency is by electric resistance heaters in the upper portion of the solar storage tank, as shown in Figure 4-2. Temperature stratification in the tank, accomplished by bringing cold water from the main into the bottom and by circulating through the collector from the bottom of the tank to the upper portion of the tank, thereby prevents auxiliary heat from increasing the temperature of the water supplied to the collector. Water returning from the collector may be brought into the tank well below the level of the resistance heater (as shown by the dashed line), to that the hot supply is always available at the thermostatted temperature. In effect, the two tanks shown in Figures 4-3 and 4-4 are combined into one, with temperature stratification providing a separation. The total amount of storage is, of course, reduced unless the one tank is increased in size. If relatively high temperature water is desired, there may be an undesirable influence of auxiliary supply on collector efficiency because of some mixing in the tank.

Although the description of the above systems refers to direct circulation of water through the collector, the same factors apply to the systems involving heat exchange with antifreeze solutions or air circulating through the collector. In all cases, auxiliary heat should be supplied downstream from the solar storage tank, regardless of whether the water itself is circulated through the collector or whether heat is exchanged between the domestic water and a solar heat transfer fluid.

LOCATION OF COLLECTORS

If the slope and orientation of a roof is suitable, the most economical location for a solar collector in a residential water heating system is on the south-facing portion of the roof. The cost of a structure to support the collector is thereby eliminated, and pipe or duct connections to the conventional hot water system are usually convenient. In new dwellings, most installations can be expected on the house roof. Even in retrofitting existing dwellings with solar water heaters, a suitable roof location can usually be provided.

If the mounting of collectors on the roof is impractical, for any of several reasons, a separate structure adjacent to the house may be used. A sloping platform supported on a suitable foundation can be the base for the collector. Pumps, storage tank, and heat exchanger, if used, can be located inside the dwelling. Effective insulation on ducts and piping must be provided, however, so that cold weather operation will not be handicapped by excessive heat losses. In cold climates, collectors in which water is directly heated must be located so that drainage of the collector and exterior piping can be dependable and effectively accomplished.

TEMPERATURE STRATIFICATION IN SOLAR HOT WATER TANK

As in a conventional hot water heater, the temperature in the upper part of a solar hot water tank will normally be considerably higher than at the bottom. The lower density of hot water permits this stratification, provided that turbulence at inlet and outlet connections is not excessive. The supply of relatively cold water from the bottom of the tank to the collector permits the collector to operate at its highest possible efficiency under the prevailing ambient conditions. With a circulation rate such that a temperature rise through the collector of 15° F to 20° F occurs, the lower part of the storage tank is furnished to the collector for maximum effectiveness. If not much hot water is withdrawn from the tank during a sunny day, the late afternoon temperature of an 80 gallon tank connected to a 40-to-50-square-foot collector may be well above 100° F - even approaching the temperature in the top of the tank. Collection efficiency thus varies throughout the day, depending not only on solar availability, but also on the temperature of water supplied to the collector from the tank bottom.

TEMPERATURE CONTROL LIMIT

In addition to the differential temperature control desirable in most solar water heating systems (which sense temperature difference between collector and storage), protection against excessive water temperature may be necessary. Several possible methods can be used. In nearly all types of systems, whether direct heating of the potable water or indirect heating through a heat exchanger, a thermostatically controlled mixing valve can be used to provide constant temperature water for household use.

Figure 4-7 illustrates one method by which this type of temperature control can be accomplished. Cold water is admitted to the hot water line immediately downstream from the auxiliary heater in sufficient proportion to secure the desired preset temperature. The solar hot water tank is allowed to reach any temperature attainable, and the auxiliary heater furnishes additional energy only when the auxiliary tank temperature drops below the thermostat set point. Maximum solar heat delivery is thus achieved, and no solar heat needs to be discarded, except that which might sometimes be delivered when the main storage (preheat) tank is at the boiling point. Any additional solar heat collected

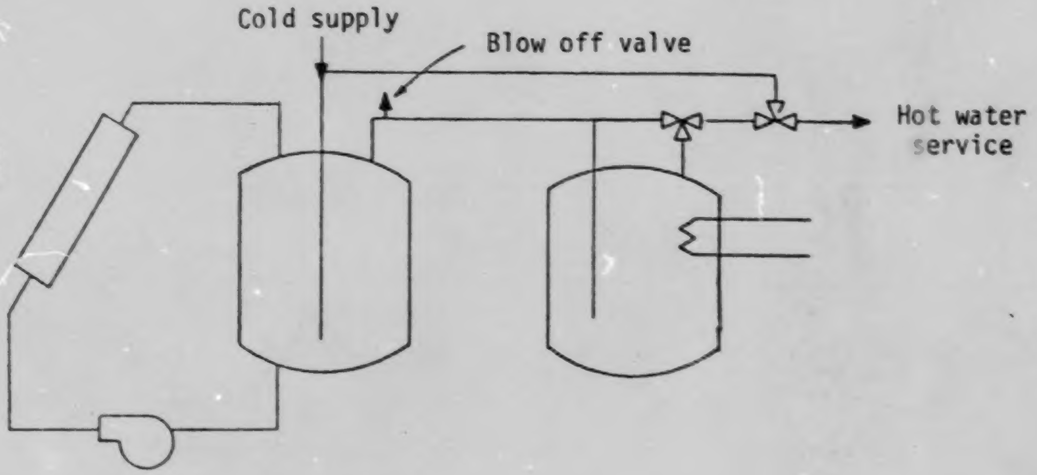


Figure 4-7. Direct Solar Water Heating with Mixing Valve.

under that condition would be dumped through a pressure relief valve, steam escaping to the surroundings. Figure 4-5 shows an optional second mixing valve for control of delivery temperature by admitting regulated amounts of solar heated water into the flow from the auxiliary heater.

A steam vent from the solar hot water system involving a dual liquid design, with heat exchange, should normally be in the hot water loop rather than the collector loop. Loss of collector fluid by vaporization is thereby avoided. It is necessary, however, in this design, that the collector tubes and associated piping be capable of withstanding pressure at least as high as developed when the steam vent valve in the storage loop is actuated. If, for example, the blow-off valve in the storage circuit is set for 50 psi, and if the collector loop containing 50 percent ethylene glycol normally operates at a temperature 20° F above the storage tank temperature, pressure in the collector loop would also be about 50 psi when the storage tank vent is actuated. (Approximate equality of pressure is due to similarity between boiling point elevation and temperature difference in the heat exchanger).

An alternative to the high pressure collector capability described above is available in the form of an organic heat transfer fluid having a high boiling point. Dowtherm J or Therminol 55 have boiling points above 300° F, so if one of these fluids is used, the development of pressure in the collector loop would not occur, even when the storage system is venting steam at 50 psi. This option appears considerably more practical than the pressurized collector required with aqueous systems if the dual-liquid design is utilized.

Still another option for high-temperature protection is available if the collector is used as a heater for a high-boiling organic liquid or for air. To prevent the storage tank from reaching a temperature higher than desired, a limiting thermostat in that tank can be used simply to discontinue circulation of the heat transfer fluid (organic liquid or air) through the collector and heat exchanger. No additional heat is, therefore, dissipated in the form of collector heat loss. The collector temperature rises substantially, frequently above 300° F, but if properly designed, the collector suffers no damage. This system is probably the safest and most dependable of those herein described. With a reliable limit switch in the storage tank, there can be no dangerous pressure developments anywhere in the system. In addition, there is no loss of water (in the form of steam) even when there is no use of hot water for long periods.

If the hot water/cold water mixing valve downstream from the auxiliary heater is not used, a temperature limit control in the solar storage tank can be set at the maximum desired temperature of service hot water. Water, therefore, cannot be delivered at any temperature higher than the set point in the solar storage tank or the set point in the auxiliary heater, whichever is higher. Less solar storage capability would be involved in this design, however, because the solar storage tank is prevented from achieving higher temperatures, even when solar energy is available.

In a direct type of solar water heater operating at service pressure, with potable water circulating through the collector, a venting valve is provided near the top of the collector. It would have to be set for release at a pressure several pounds higher than the maximum in the service supply, so the collector storage system must withstand pressure usually above 50 psi. Occasional water loss through venting of steam would be expected.

If a non-pressurized direct type of solar water heater is used, with a float valve in the storage tank, the pressure relief valve can be set to operate at a pressure only slightly above atmospheric. Alternatively, the collector or storage tank may be continuosly vented. Oversupply or under-use of solar heated water results in boiling and venting of the storage tank.

PERFORMANCE OF TYPICAL SYSTEMS

GENERAL REQUIREMENTS

A typical family of four persons requires, in the United States, about 80 gallons of hot water per day. At a customary supply temperature of about 140° F, the amount of heat required, if the cold inlet is at 60° F, is about 50,000 Btu per day.

There is a wide variation in the solar availability from region to region and from season to season in a particular location. There are also the short-term radiation fluctuations due to cloudiness and the day-night cycle.

Seasonal variations in solar availability result in a 200 to 400 percent difference in the solar heat supply to a hot water system. In the winter, for example, an average recovery of 40 percent of 1200 Btu of solar energy per square foot of sloping surface would require approximately 100 square feet of collector for the 50,000 Btu average daily requirement. Such a design would provide essentially all of the hot water needs on an average winter day, but would fall short on days of less than average sunshine. By contrast, a 50-percent recovery of an average summer radiant supply of 2000 Btu per square foot would involve the need for only 50 square feet of collector for satisfying the average hot water requirements.

It is evident that if a 50-square-foot collector were installed, it could supply the major part, perhaps nearly all, of the summer hot water requirements, but it could supply less than half the winter needs. If, on the other hand, a 100-square-foot collector were employed in order that winter needs could be more nearly met, the system would be oversized for summer operation and excess solar heat would have to be wasted. In such circumstances, if an aqueous collection medium were used, boiling of the system would occur and collector or storage venting of steam would have to be provided.

The more important disadvantage of the oversized collector (for summer operation) is the economic penalty associated with investment in a collector which is not fully utilized. Although the cost of the 100-square-foot collector would be approximately double that of the 50-square-foot unit, its annual useful heat delivery would be considerably less than double. It would, of course, deliver about twice as much heat in the winter season, when nearly all of it could be used, but in the other seasons, particularly in summer, heat overflow would occur. The net effect of these factors is a lower economic return, per unit of investment, by the larger system. Stated another way, more Btu per dollar of investment (hence cheaper solar heat) can be delivered by the smaller system.

As a conclusion to the above example, practical design of solar water heaters should be based on desired hot water output in the sunniest months rather than at some other time of year. If based on average daily radiation in the sunniest months, the unit will be slightly oversized and a small amount of heat will be wasted on days of maximum solar input. And quite naturally, on partly cloudy days during the season, some auxiliary heat must be provided. In the month of lowest average solar energy delivery, typically one-half to one-third as much solar heated water can be supplied, or actually the same quantity of water but with a temperature increase above inlet only one-half to one-third as high. Thus, fuel requirements for increasing the temperature of solar heated water to the desired (thermostatted) level could involve one-half to two-thirds of the total energy needed for hot water heating in a midwinter month.

Sizing the Collectors

The curves shown in Figure 4-8 may be used to estimate the solar collector size required for hot water service in residential buildings having typical hot water systems. The system is assumed to be pumped liquid type, with liquid-to-liquid heat exchange, delivering hot water to scheduled residential uses from 6:00 a.m. until midnight. The shaded band represents results of computer calculations for eleven different locations in the United States. The cities included in the study are Boulder, Colorado; Albuquerque, New Mexico; Madison, Wisconsin; Boston, Massachusetts; Oak Ridge, Tennessee; Albany, New York; Manhattan, Kansas; Gainesville, Florida; Santa Maria, California; St. Cloud, Minnesota; and Washington, D. C. The hot water loads used in the computations range from 50 gallons per day (gpd) to 2000 gpd. The sizing curves are approximate and should not be expected to yield results closer than 10 percent of actual value.

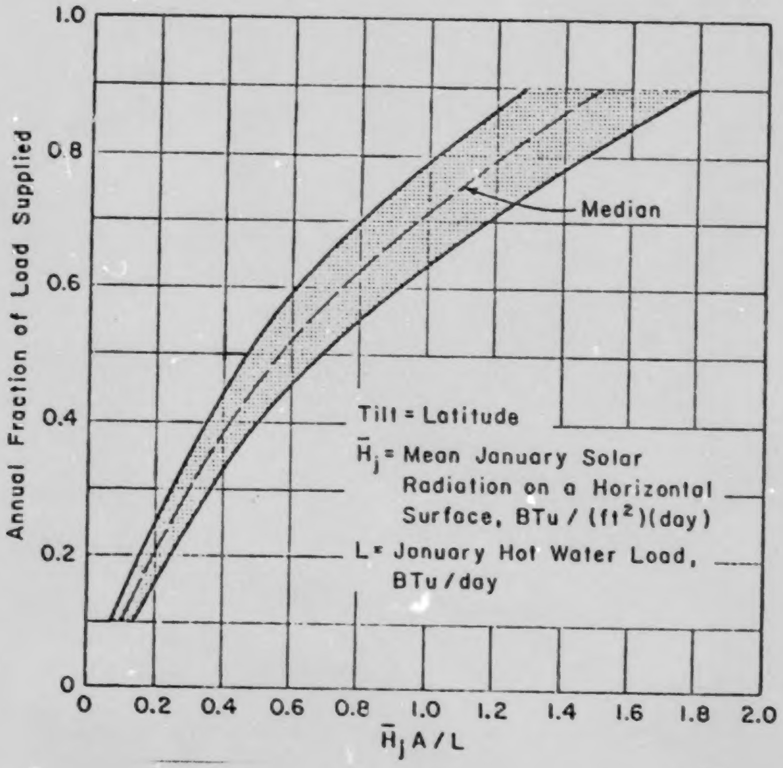


Figure 4-8. Fraction of Annual load supplied by Solar as a Function of January Conditions for Hot Water Heaters

The vertical axis shows the fraction of the annual water heating load supplied by solar. The horizontal axis shows values of the parameters, $\bar{H}_J A/L$, which involves the average daily January radiation on a horizontal surface, $\bar{H}_J$; the required collector area, A, to supply a certain percentage of the daily hot water load, L. The January average daily total radiation at locations in the United States can be estimated from the radiation map in Figure 4-10. Values on the map are given in Btu/(ft^2)(day). The curves are not applicable for values of greater than 0.9.

It should be remembered that the service hot water load will be nearly constant throughout the year, while the solar energy collected will vary from season to season. A system sized for January, with collectors tilted at the latitude angle, will deliver high temperature water and may even cause boiling in the summer. On the other hand, a system sized to meet the load in July will not provide all of the load in the winter months. Orientation of the collector can partially overcome month-to-month fluctuations in radiation and temperature.

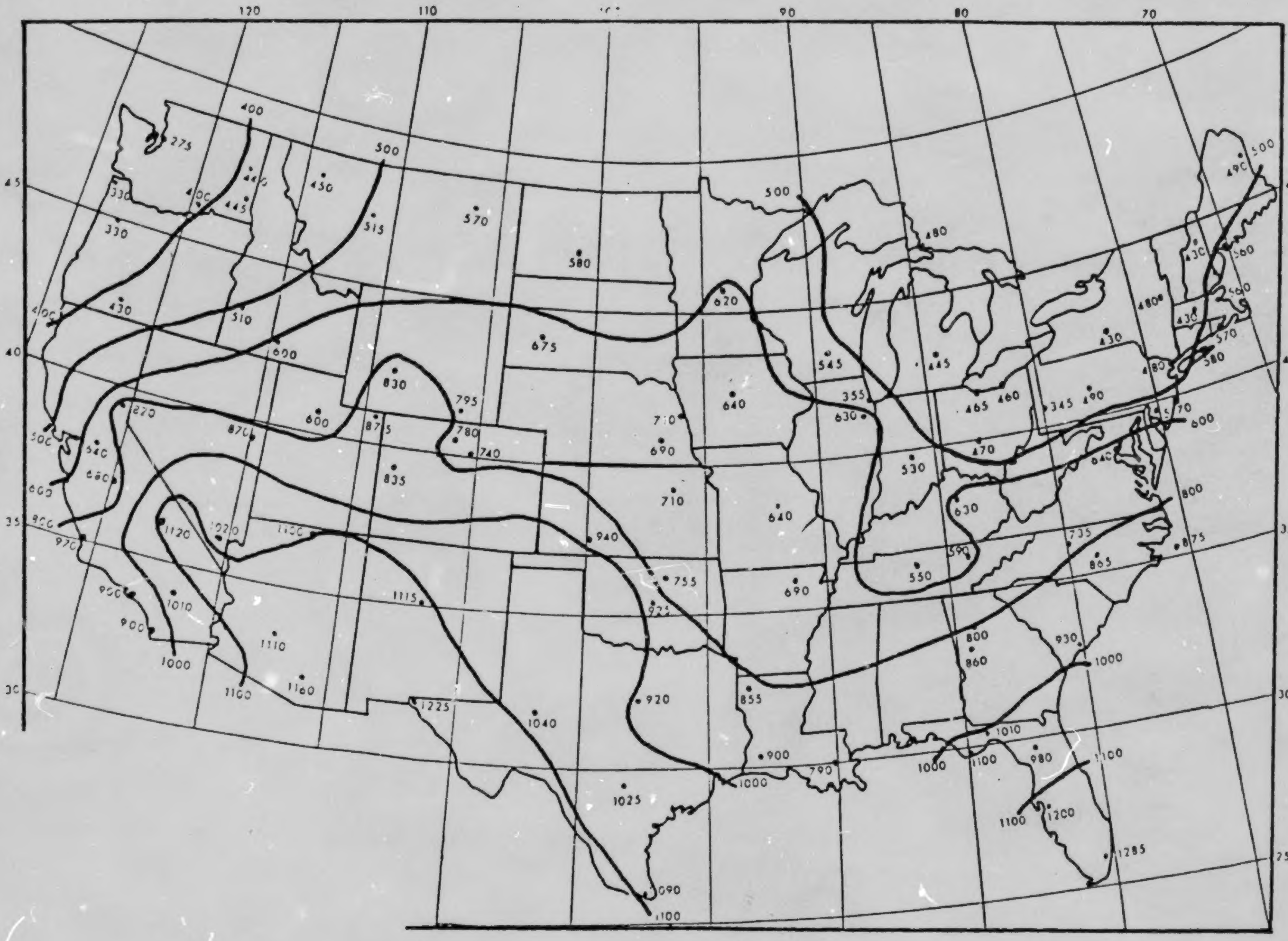
Sizing Examples

Example 1. Determine the approximate size of collector needed to provide hot water for a family of four in a residential building in Kansas City, Missouri.

Solution: The average daily service hot water load in January is:

$$L = 80 \text{ gallons/day} \times 8.33 \text{ pounds/gallon} \times 1 \text{ Btu/(lb)}(^\circ\text{F}) \times (140^\circ \text{ F} - 50^\circ \text{ F}) = 60{,}048 \text{ Btu/day}$$

The desired service water temperature is 140° F and the temperature of the cold water from the main is 50° F. The total average solar radiation, $\bar{H}_J$, available in January, from Figure 4-9, is 680 Btu per square foot per day. For a water system to provide 60 percent of the annual load, from Figure 4-8, $\bar{H}_J A/L$ is about 0.8. Therefore: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930
1000
855
920
1225
1040
1010
900
790
1000
1000
1100
980
1100
1025
1100
1200
1285
1090
1100

Figure 4-9. Average Daily Solar Radiation (Btu/ft^2), Month of January

$$A = 0.8 \times L/\bar{H}_j = (0.8 \times 60048)/680 = 70.6 \text{ square feet.}$$

If 3-by-8-foot collector modules are available, 2.9 units would be required. Three collector units should therefore be used.

Example 2. Determine the size of collector needed to provide hot water for a family of four in Albuquerque, New Mexico.

Solution: The monthly load will be approximately the same as in Example 1:

$L = 60{,}048$ Btu

From Figure 4-9, $\bar{H}_j = 1115$ Btu/(ft^2)(day). For a system to provide 60 percent of the annual load, Figure 4-8 shows that $\bar{H}_j A/L$ is approximately 0.8. The collector area required is:

$A = (0.8 \times 60048)/1115 = 41.8.$

Using 3-by-6-foot collector modules, 2.3 units would be required for this system, either two or three modules should be used. If two modules are used, the system would be expected to provide less than 60 percent of the annual load.

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF THE COLLECTOR ARRAY - MANUAL METHOD

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF THE COLLECTOR ARRAY - MANUAL METHOD

Preliminary Considerations

Before the design and sizing process begins, it is essential to make a quick analysis of the application and site, using the parameters outlined in the first module of this course. A review of system choices would be helpful at this point, particularly those problems that may be inhenent in the design.

Overheating

At high temperatures, materials such as insulation, absorber plate coatings, and sealants may break down and structural components of the collector may undergo dimensional changes. When the collectors are not being cooled by the transfer fluid, stagnation temperatures can be as high as 300 to 400 degrees Fahrenheit.

Outgassing

One common overheat problem is outgassing, which occurs as volatile materials (coatings, sealants, binders, insulation) "boil off", depositing a film when "condensing" against the cooler surface of the collector glazing. This deterioration may affect collector performance. Degradation of sealants also increases moisture related problems. Most manufacturers are now addressing this problem in their selection of there materials.

Charring

Some collector arrays have incorporated plywood or wood components, either in the collector construction or the collector installation. Constant or long-term exposure to overheat temperatures can reduce the moisture content of the wood, weakening the wood and reducing its flash temperature. Exposure of the charring component to sufficient air may cause a fire. One liquid system experienced charring and delamination of plywood components of the collectors and the roof. Collector overheating initially caused internal insulation to deteriorate, which first exposed the plywood in the collector to charring temperatures, and then exposed the plywood of the roof to similar temperatures.

While manufacturers work to correct this problem, wood building components comprising the collector's collector installation or adjacent roof structures should be avoided or be protected from high temperatures. A similar condition may occur with some foam insulations.

Boil-out

Liquid collectors may become damaged when high stagnation temperatures cause either water or hydrocarbon based fluid to boil. "Steam" from the fluid increases internal pressures. This may open leaks in the collectors, hose or piping connections, cause separation in the absorber plate, or literally blow out the glazing.

Overheat Deterioration

High stagnation temperatures may also exceed the anticipated or designed operating range for materials and fluids in the collectors. High temperature may cause chemical decomposition of corrosion-inhibiting or anti-freeze solutions, which would lead to corrosion build up and leakage if not noticed and corrected. Insulation can char, melt, or shrink at high temperatures, increasing the potential for damage to other heat susceptible materials. Standard materials such as 50/50 solder weaken at high temperatures, increasing the potential for damage to piping and collector connections, and collector plate joints from thermal expansion. Unanticipated high temperatures may also cause plastic glazing to melt or sag, reducing collector efficiency.

Excess Heat Disposal

A collector array designed for space heating and water heating, or one combining solar cooling may be under-utilized during certain seasons, and a means to prevent or dispose of excess heat must be incorporated. Stagnation may also occur during installation or before initial operation. One method of handling this problem of excess heat is a heat dump - a component or mode of operation designed to lose or reject heat from the system. Another method is simply shading the collectors during periods of anticipated overheat. More manufacturers are now designing collectors to withstand the effects of overheat, which is a good approach. However, as more collectors withstand stagnation, other overheat problems in other subsystem components cannot be forgotten or underrated.

Insulation

Insulation as a standard collector component will minimize loss of collected solar heat prior to transfer to storage or load. Insulation of the piping, ducting, and other connections in the collector array will also minimize losses. Though temperature differentials between outside air and operating fluids are still considerable, many systems in warmer climates overlook the benefit of sufficient insulation during design and installation, and soon retrofit insulation.

Passive systems lack the collector insulation normally installed as a component of active system collectors. Several passive systems have incorporated operable shutters or operable insulation systems to minimize loss of collected solar heat from storage and conditioned space during periods of low insolation and at night. Experience with manually operated systems has shown that operational difficulties may reduce the effectiveness of the overall solar system if the insulation is not properly placed and removed. One manual system had a lower net solar system effectiveness that if no insulating shutters had been used.

Wind Speed

For structural and thermal considerations, exposed collectors may need protection from the wind and may be shielded for protection. Many tracking, concentrating liquid collectors are placed in a "stowed" position during periods of high wind or insufficient sun. The time required by these systems to reposition the reflectors reduces the available solar and may reduce operational efficiencies. Another liquid system array was placed on the ground, rather than the roof, to reduce the thermal losses in the higher speed winds at the roof level.

Reflective Surfaces

Reflective surfaces adjacent to glazing or storage increase the incident solar at the point of collection. One system surrounded a passive domestic hot water tank with reflective surfaces inside a glazed "closet" which, combined with diurnal automatic collector insulation and evening hot water use, contributed to a 25 percent solar fraction for an auxiliary water heating application. Another system used reflective surfaces below roof clerestory glazing, contributing to an 80% solar fraction for a space heating application. Reflectors may also be placed in front of active collectors.

Service Access

The location and positioning of collectors should anticipate the need for easy access for regular maintenance and periodic service. Non-integral collectors and collectors with easy to remove glazing are most accessible.

Workmanship

A collector system with fewer field connections will provide the fewest potential leaks. Manufacturer's quality control procedures and inspections cannot be adequately duplicated in the field. Although the level of workmanship required for a liquid collector system is more common in construction trades that that required for an air collector system, both types of systems can experience problems which result from inexperienced supervision and simple oversight.

Corrosion

The combination of components of various metals, such as aluminum, copper and iron, in contact with the collector fluid can cause corrosion of couplings or connectors and scaling or deposits in piping and absorber plates, among other detrimental occurences. Corrosion may lead to leaks which cause collector overheating, require replacement of components, or lead to dilution of anti-freeze concentration in automatic-feed water fluids which can cause collector freezing. In addition to causing reduced performance, scaling and iron deposits may lead to imbalances in the fluid flow which again cause collector overheating, prevent complete freeze protection for drain-down liquid systems, and require replacement of piping, absorber plates, heat exchangers, and even storage tanks.

In addition, the use of some anti-freeze and corrosion inhibitor solutions may accelerate the deterioration of other system components, such as the storage tank heat exchanger and storage tank lining, if these solutions are not rated for overheat temperatures.

The materials and fluids chosen for the collector system must be chemically compatible and able to withstand anticipated operating and stagnation temperatures.

Performance Effects

The effect of minor air leakage on the thermal performance of solar heating systems has not been adequately quantified through instrumented site-monitoring, although serious modeling approaches have been undertaken. Nevertheless, air leaking into or out of a solar air heating system may typically be 30 to 40% of the design air flow in systems. and result in increased auxiliary system energy use.

Pressurization of Air Systems

The direction of air collector leakage is largely determined by the location of the collector fan relative to the collectors. Systems in which the collector fan is downstream of the collectors normally demonstrate a slight negative pressure in the collectors relative to atmospheric, tending to draw air into the collector from outside. Systems with the collector fan upstream from the collectors tend to force air out of the collectors.

An air system with slightly negative pressure, rather than positive pressure, in the collectors is a preferred design for at least two reasons. First, a system with negative pressure pressurizes the conditioned space positively, reversing the direction and reducing the level of normal infiltration/exfiltration of outside air into the conditioned space. Reduced infiltration reduces heating loads. Second, a system with negative pressure draws preheated outside air into the system through the collectors, and a system with positive pressure exhausts warmer air from storage through the collectors. The former system conserves net energy.

The magnitude of these leakages in either direction has also been shown to be dependent on the system operating mode at any given time. Design of collector joints, connections and seals to standards consistent with various levels of pressurization can reduce collector leakage.

LIGHTNING PROTECTION

Solar systems are usually installed on buildings that are sited some distance from taller buildings and trees which could shade collectors. Many solar systems are also installed in rural locations. And in most cases, materials and optimum solar handling invite lightning. Under the circumstances, normal system grounding through the water or fluid piping, or electrical circuits may not be sufficient protection to prevent damamge to the collectors and other system components and to reduce the potential for building fire. A separate lightning rod or collector ground may be advisable.

Design Problems

After considering the potential design problem inherent in the component, let us look at some of the problems that may occur as a result of design choices made because of installation limitations and overprojection of performance as stated in "Solar Design and Installation Experience" from the National Solar Data Network (D. O. E. Publication).

This section describes the thermal performance of arrays of solar collector panels on operating solar energy systems in the National Solar Data Network. The assessment of collector array performance is a critical aspect of the design of active solar systems. System designers usually use the single panel efficiency test results derived from the criteria of ASHRAE Standard 93-77 as an input to their analysis of predicted collector performance. However, the ASHRAE 93-77 test results are only indicative of collector thermal performance under a narrow range of operating conditions. Arrays of collector panels in operational solar energy systems are exposed to a wide range of system and environmental conditions which affect their efficiency.

The major operational considerations of collector arrays which may lead to differences from single-panel test results are:

1. incidence angle variations,
2. fraction of diffuse radiation,
3. dust on collector covers,
4. higher wind speeds,
5. imbalances in the fluid flow distribution in arrays,

6. corrosion and fluid scaling,

7. different ranges of outdoor ambient temperature and fluid temperatures in the collectors, and

8. lower or higher fluid flow rate per panel than recommended.

For this report, the effects of these individual environmental and operational conditions on the efficiency of collector arrays have not been isolated, but indicate the performance of arrays under typical operating conditions related to single-array standard test performance. Most arrays are subject to a combination of these conditions in varying degrees.

An example of collector array performance is presented to show the relationship of the efficiency of an array under actual dynamic operating conditions to the single-panel, quasi-steady-state ASHRAE test conditions. Figure 5-1 is a collector array efficiency plot of a liquid flat-plate system located in Berkeley, California, for October, 1980. This system is installed so that all modules are connected in parallel with four rows of 17 modules in each row.

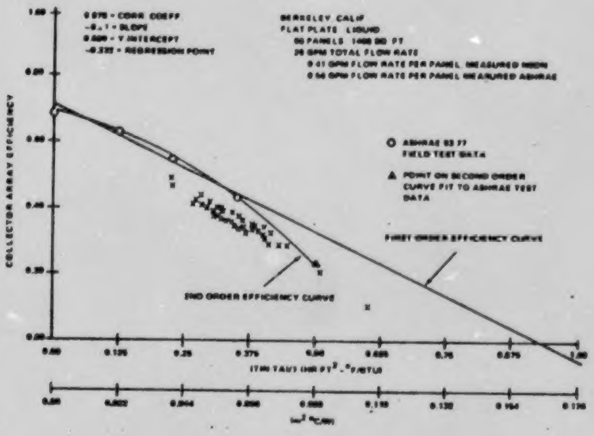


Figure 5-1: Average Collector Efficiency
October, 1980, Berkeley, California

The whole array is tilted at 45 degrees from the horizontal with an azimuth angle of zero degrees (south). The collector absorber plate is copper with black chrome selective surface, and the modules are single glazed with low iron glass. The plotted collector array efficiencies are hourly averages as are the values of (TIN-TA)/I as compared to instantaneous values recorded by ASHRAE testing.

Conclusions drawn about the difference between field-measured array efficiency and ASHRAE 93-77 test results may vary greatly depending on whether or not the data is fitted by the second order polynominal or the straight line. In Figure 5-2, the second order fit to the ASHRAE data yields efficiencies which are somewhat higher than the first order curve fit efficiencies of the ASHRAE data low and mid-range values of (TIN-TA)/I. However, for higher range values of (TIN-TA)/I, the second order curve fit efficiencies are significantly lower.

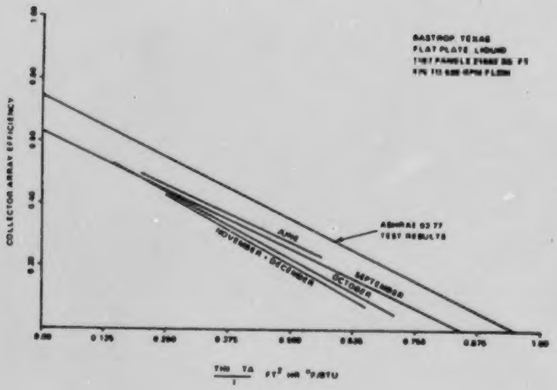


Figure 5-2: GSA Federal Youth Center -
Collector Efficiencies 1980

In a comparison to the first order fit of the ASHRAE data, the efficiency of the array would be lower than the single-panel test results through the complete range of values of (TIN-TA)/I. However, as is seen in Figure 5-1, when a second order polynominal is fitted to the ASHRAE test data, the array efficiencies for mid and low range values of (TIN-TA)/I but about equal to the ASHRAE efficiencies for higher values of (TIN-TA)/I. The hours with higher values of (TIN-TA)/I in the array field data result from lower values of solar radiation, I. The efficiencies with high values of (TIN-TA)/I derived from ASHRAE testing are a result of higher inlet fluid temperature to the collector. Since the difference between inlet fluid temperature and ambient air temperature is greater for the high value of (TIN-TA)/I in the ASHRAE test, the heat losses will be higher in the ASHRAE TEST, thus resulting in a reduction in efficiency. Differing meteorological conditions are also a significant factor in the efficiency testing of solar collectors. The ASHRAE 93-77 testing for these collectors was performed in a region of clear, dry air masses and high insolation levels. The actual weather conditions at most sites will vary considerably with time. The Berkeley, California example includes much hazier conditions (both pollution and water vapor hazes), and greater amounts of cloud cover and fog. Thus, the overall average percentage of diffuse radiation may be higher and result in less favorable conditions for actual solar collector performance.

It must be noted that collector arrays include headers and piping while single collector panel tests do not. The additional losses reduce array performance, although good insulation and manifold design can minimize the effects. In buildings with long piping runs, the losses may be significant.

An efficiency curve for a collector array based on only one month of data does not characterize the annual performance of the array since there are often significant differences in seasonal performance. In Figure 5-2, the collector array performance of an NSDN site in Bastrop, Texas for the period June, 1980 through December, 1980 is presented. The lines drawn are the least square fits to the hourly data for each month. It may be seen that the slope of the lines increases as the weather conditions and solar angles change from June through December. Since the array is tilted at 25 degrees from the horizontal and 30 degrees west of south, the array orientation is optimized for the higher solar elevation angles of the summer. Also, the percentage of diffuse radiation increased from the summer through the winter, since the summer of 1980 in Texas was extremely hot and dry and the months of December and November, 1980, were considerably more cloudy than the 30-year normals for the region. An additional factor in the reduced efficiencies of October, November, and December was the increasingly lower outdoor ambient temperature which increased the heat loss from the collectors when the difference between collector fluid temperature and outdoor ambient temperature increased.

The piping to the subarrays in this system of 1,187 collector panels is copper, and the main trunk line off which branch the lines to the subarrays is a steel pipe. This condition leads to the formation of scale in the system. Before the system was put into operation, the collector array was in stagnation for two years. Thus, scale in the system and larger incidence angles undoubtedly accounts for some of the significant reductions in performance from the ASHRAE 93-77 test results.

Figures 5-3 and 5-4 show collector array efficiency plots for high efficiency arrays and low efficiency arrays respectively. Low efficiency arrays are defined here as arrays where the efficiency is significantly lower than ASHRAE test results throughout most of the range of operating points of the system. However, if most of the operating points are concentrated at the lowest values of (TIN-TA)/I, the overall efficiency of the collector array can be good. All of the high efficiency arrays are single-glazed, flat-plate collectors using water as the heat transfer fluid. Four of these arrays are draindown systems and the other two are in warm climates where freeze protection is not necessary. All of the low efficiency collector arrays except for two are flat-plate collectors using glycol or oil as the collector heat transfer fluid. One is a site-built collector array (Terrell E. Moseley) with water draindown for freeze protection. This site-built array efficiency curve has the steepest slope of all the lines in Figures 5-3 and 5-4, indicating that it has the highest loss coefficient of this sample of sites. However, the overall efficiency of this array is good at 30%, based on total insolation, since most of the system operation occurred at low values of (TIN-TA)/I. These low values of operating point are due to low storage tank temperature, which results in low inlet temperatures to the collectors.

It is difficult to compare these collector arrays based on the type of heat transfer fluid or any other factor, since there are many other variables which affect the performance of collector arrays. Some of the differences between arrays are the manifolding of the collectors, which affects the balancing of the flow to the individual panels within the array, the different materials and designs of the sites in different parts of the country, and the flow rate per unit area of collector surface. For nearly all of the NSDN sites, the flow rate per unit area of collector area was high enough so that efficiency would not be significantly degraded under even flow distribution conditions. One of the systems is a concentrating collector array with tracking absorber tubes, using water as the heat transfer fluid. This site, Reedy Creek, has the shallowest slope of all the efficiency curves due to the collector tracking,

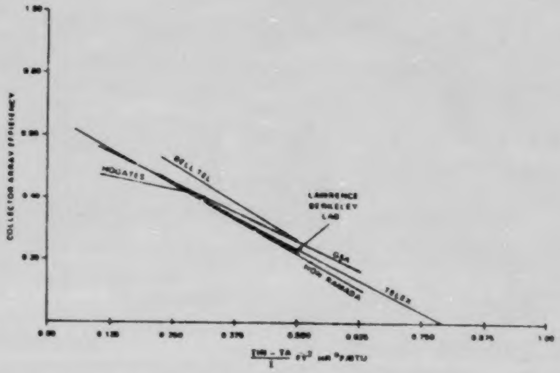


Figure 5-3: High Efficiency Collector Arrays (All are Flat-Plate, Single-Glazed Using Water as the Heat Transfer Fluid)

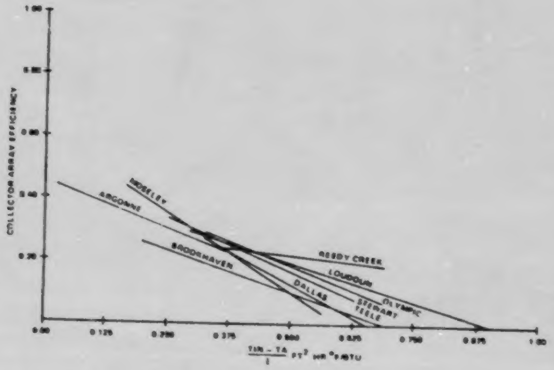


Figure 5-4: Low Efficiency Collector Arrays

which optimizes the angle between incident radiation and the collector absorber. All-day collection subsystem efficiencies, based both on total solar radiation incident and radiation incident during collector pump operation, vary widely between sites. This is because collection efficiency is a function of collector type, storage temperature, and load patterns.

Figure 5-5 is a graph of the efficiency of 17 sites for the winter heating season of 1979-1980. Two different collector array efficiency calculations are performed:

1. based on the total solar radiation incident in the plane of the array, and

2. based on solar radiation that is incident during times when there is active fluid flow through the collectors.

The mean total collection efficiency for these sites was 24%, with a standard deviation of ± 6%. However, the operational mean collection efficiency was much higher at 36%, indicating significant threshold losses for many of the solar energy systems. Threshold losses are primarily a function of average storage temperature, as well as collector and control system design. Low average storage temperatures, low loss collector arrays, and accurate and sensitive control systems all tend to reduce the amount of threshold loss.

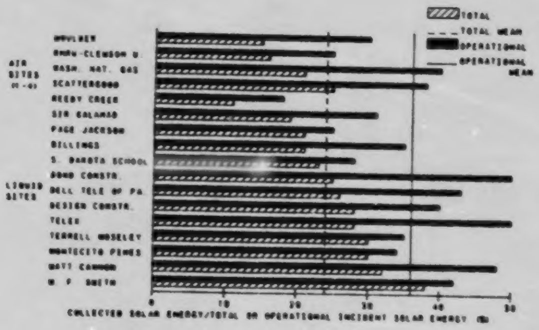


Figure 5-5: Collector Array Efficiency, Total and Operational

The M. F. Smith site had the highest solar collection efficiency. This is due to the low average storage temperature of 58° F which, as a result, allows the system to collect during times of low solar radiation intensity. This site has a solar-assisted heat pump which allows it to operate at that temperature level. The other heat pump sites fall, in general, above the mean in terms of total collection efficiency.

The Reedy Creek site utilizes concentrating collectors to supply solar energy to an absorption chiller. The combination of a high average storage temperature of 164° F, problems with maintaining accurate tracking of the collectors, and the inability of concentrating collectors to collect a significant portion of the diffuse radiation, lead to the efficiency at this site falling below the mean. For the eight sites below the mean efficiency based on total insolation, the average storage temperature was 112° F. For the remaining sites above the mean, the average storage temperature was 102° F. Ranges of these NSDN collection efficiencies are 14%-30% for water draindown systems, 19%-25% for glycol systems, and 15%-25% for air systems. Thus, the range of efficiencies are similar for these three types of systems.

Ranges of operational collection efficiencies, based on solar radiation incident during pump operation, are 25%-50% for water draindown systems, 23%-50% for glycol systems, and 26%-40% for air systems. Significant differences often occur between total collection and operational collection efficiencies due to control system problems in some cases, and high storage tank temperatures in some other cases. When storage tank temperature is high, there is a higher threshold level of radiation intensity, which must be reached before the collector absorber is hot enough to add energy to the storage medium.

The amount of thermal energy collected per unit of electrical operating energy required to collect the energy is an important factor to be considered in the design of a collector subsystem. Collector subsystem coefficient of performance (COP) is defined as the amount of useful energy gained divided by the amount of electrical operating energy required to run the pumps or fans. Ranges of collector subsystem COPs were from 12-43 for water draindown sites, 15-37 for glycol loop systems, and 17-30 for air systems. One site required no operating energy for collection since it is a water thermosiphon system. This type of system is limited to nonfreezing climates. The reasons for widely varying collection COPs are large differences in the efficiency of collectors and different sizes of pumps, which are related to the lengths and resistance to flow of manifolding and piping runs. The sizing of a pump or fan is also dependent on the designed flow rate per square foot of collector of the system. The average collection COP of the air systems was lower than for the liquid systems. This is due in part to the large amount of operating energy required to force air through high resistance rock bed storage units. Also, fans require more specific operating energy per unit mass of air moved than pumps per unit mass of fluid moved. In a water draindown system, which normally uses pumps only on the collection side of storage, there is the potential for higher collection COPs. Anti-freeze systems, which have an additional pump on the storage side of the collector subsystem heat exchanger would tend to use more operating energy. The need for a heat exchanger may be eliminated in most draindown systems. Coupled with the superior overall heat transfer properties of water, thermal efficiency would be increased. Water has a lower viscosity, higher thermal conductivity, and higher specific heat than glycol solutions, silicon oils, or hydrocarbon oils. Water draindown systems, however, could be subject to increased corrosion due to the exposure of the system to air. Care must be taken in the design of draindown systems to make sure there are no undrained pockets which could freeze. Overall, NSDN data has shown that active collection systems using air heating solar collectors with rock storage, and liquid heating solar collectors with water storage, can perform with similar efficiencies. Individual array efficiencies ranged from performance near to ASHRAE 93-77 tests to efficiencies much below single-panel test results. The major problems leading to poor performance are inadequate flow distribution, degradation of collector characteristics due to improper design, installation and operational considerations, and high rates of air leakage in air systems.

In addition to the evaluation of collector array performance, a common measure of the effectiveness of a solar system is related to the amount of solar energy delivered to the load for each square foot of collector area. Figure 5-6 identifies this parameter for the sites indicated and shows averages for each of the major categories; DHW, heating; etc. Although the averages are of some value in identifying trends, it can be seen that these results are also very much site specific. The amount of solar energy collected and used is frequently as much a function of the temperature set points, operating control strategy, and system load requirements as of the collector array design itself.

In summary, some of the major considerations in the design of action solar collector arrays are:

1. Fluid flow distribution to all collector modules within an array should be evenly balanced to ensure high efficiency for all modules.

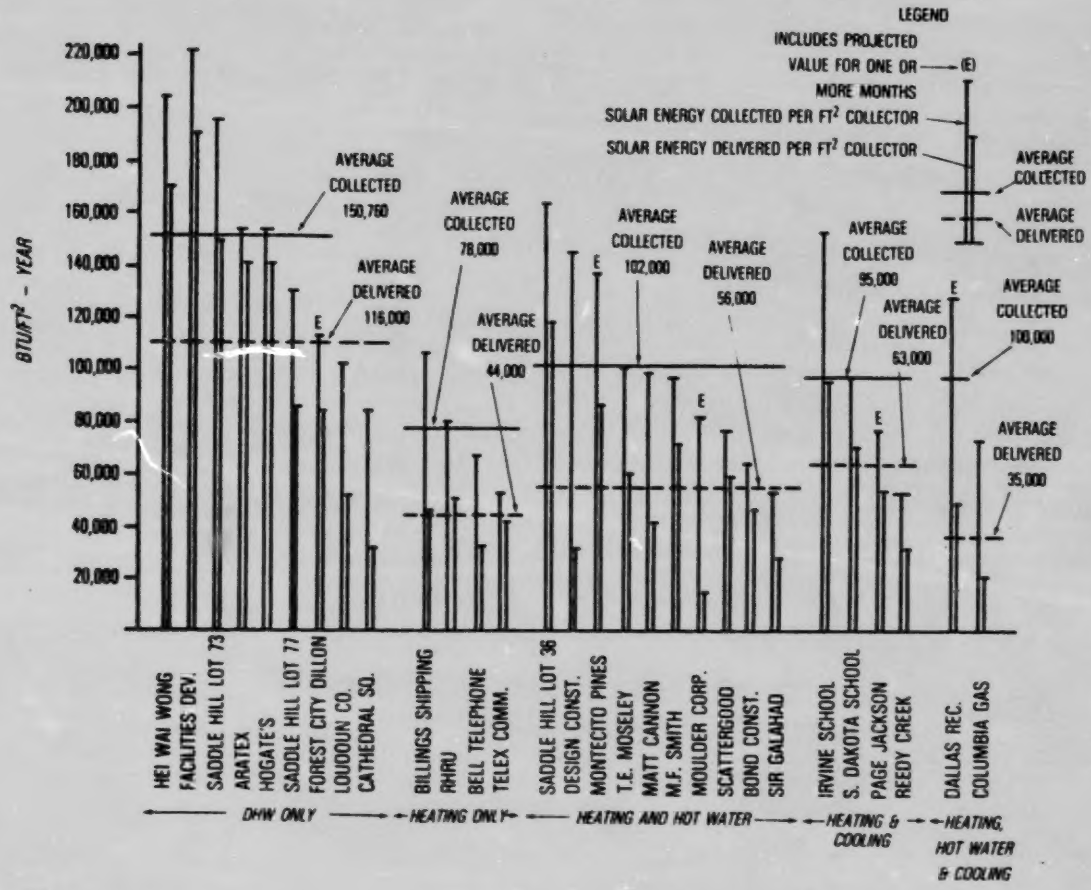


Figure 5-6: Solar Energy Collection and use, Btu/ft^2 - Season, for Selected NSDN Sites, 1979-1980.

2. Consideration of corrosion and scaling problems should be foremost in the selection of materials and heat transfer fluids for solar energy systems.

3. The climate of the site and how it differs from the conditions of ASHRAE 93-77 testing should be recognized when estimating the useful energy collection. Wind, percentage diffuse radiation, ambient temperature, and insolation levels need to be considered as they very with the seasons. Simulations of collector performance should account for as many of these meterological variables as possible.

4. A major goal in the design of active solar collection subsystems is to minimize the use of electrical energy to operate the system. Electrical operating energy, while sometimes overlooked, is often significant in magnitude, and detracts from the eventual energy savings.

The review material presented in the previous pages should be viewed as areas of concern, but the problems discussed are not insurmountable. Keep them in mind as the design process for the collector array continues.

Sizing Solar Collectors

The initial step in sizing the various components in a solar heating system is to size the collector. Because collector Btu output and system utilization of these Btu's is dependent upon a number of weather and component variables, a great deal of research has been conducted to study system performance and to develop practical collector sizing procedures. Presently, there are several sizing procedures proposed and used throughout the industry. These vary from simple rules of thumb, such as the number of square feet of collector as a fraction of the square feet of the home (e. g., collector area equals from 1/3 to 1/2 the floor area), to highly sophisticated computer programs. Many collector manufacturers have selected a particular procedure and refined it for use with their respective proprietary collectors.

For example, Table 5-1 shows a tabulation of the output of a specific manufacturer's collector panel in thousands of Btu's for each month of the winter heating season for selected cities in the United States. The table indicates that in January, for the city of Columbus, Ohio, a single collector panel from this manufacturer will supply 280,200 Btu's. Armed with this information and the monthly energy needs of the structure based on building heat loss and monthly degree days, the number of panels required to supply some fixed percentage of the heating requirements can be determined.

MONTHLY OUTPUT,/COLLECTOR PANEL — MBTU

°Latt. North	Location	Collector Tilt	October	November	December	January	February	March	April
48.2	Glasgow, Montana	60°	496.2	338.8	264.3	333.6	436.7	536.7	450.5
43.6	Boise, Idaho	55°	543.2	387.4	288.3	313.1	404.4	494.4	481.9
40.0	Columbus, Ohio	55°	420.0	257.9	259.3	280.2	312.1	394.0	360.8
35.4	Oklahoma City, Oklahoma	50°	637.3	554.2	508.1	504.6	494.2	556.3	481.9
40	Salt Lake City, Utah	55°	536.6	407.4	392.2	345.4	406.6	475.3	440.4
29.5	San Antonio, Texas	40°	660.6	537.6	497.3	527.5	529.7	590.8	484.4
32.8	Fort Worth, Texas	45°	668.8	579.8	498.6	488.8	497.4	598.6	511.6
40.3	Grand Lake, Colorado	55°	509.9	386.4	370.4	390.8	450.9	410.5	415.0
42.4	Boston, Massachusetts	55°	425.4	307.6	279.1	303.0	319.2	389.7	339.8
27.9	Tampa, Florida	40°	660.0	656.6	610.0	646.7	608.0	670.0	578.0
33.4	Phoenix, Arizona	45°	777.0	663.0	589.7	606.2	655.7	756.0	678.0
33.7	Atlanta, Georgia	45°	566.3	489.5	422.7	423.8	441.7	518.0	499.0
35.1	Albuquerque, New Mexico	50°	719.2	628.7	580.8	604.9	592.9	682.3	588.0
40.8	State College, Pennsylvania	58°	468.6	307.5	255.8	280.6	312.9	405.8	375.5
42.8	Schenectady, New York	55°	381.8	242.1	315.1	281.7	317.9	365.2	319.5
43.1	Madison, Wisconsin	55°	465.5	306.7	293.9	321.0	343.5	442.6	370.2
33.9	Los Angeles, California	50°	604.3	577.2	540.4	535.4	556.8	633.0	482.9
45.6	St. Cloud, Minnesota	60°	419.8	309.3	274.4	362.7	416.3	482.6	381.6
36.1	Greensboro, North Carolina	50°	537.8	465.1	396.0	409.5	430.2	481.4	456.4
36.1	Nashville, Tennessee	50°	556.0	424.0	354.5	325.7	380.7	456.8	438.2
39.0	Columbia, Missouri	50°	559.1	438.7	339.3	356.7	390.9	485.7	440.9
30.0	New Orleans, Louisiana	40°	589.7	506.8	390.5	415.8	396.6	473.5	448.2
32.5	Shreveport, Louisiana	45°	590.4	475.3	419.6	459.6	452.0	534.7	461.1
42.0	Ames, Iowa	55°	378.4	262.9	197.7	238.2	294.9	368.2	354.7
42.4	Medford, Oregon	55°	455.1	298.4	213.4	255.8	343.4	434.1	412.3
44.2	Rapid City, South Dakota	60°	550.9	436.3	383.3	405.1	453.2	518.0	420.3
38.6	Davis, California	50°	719.3	536.9	401.3	448.9	515.6	666.1	647.2
38.0	Lexington, Kentucky	50°	616.5	477.4	377.8	359.6	411.1	487.5	479.9
42.7	East Lansing, Michigan	55°	415.0	261.7	230.1	247.5	314.2	387.4	313.4
40.5	New York, New York	55°	505.8	389.4	328.0	357.1	396.6	451.9	381.8
41.7	Lemont, Illinois	55°	477.1	352.7	321.4	343.6	373.9	450.3	365.4
46.8	Bismark, North Dakota	60°	491.1	346.6	279.9	335.6	405.7	476.6	411.9
39.3	Ely, Nevada	55°	636.7	559.0	475.8	481.1	512.4	597.5	483.4
31.9	Midland, Texas	45°	651.5	596.2	540.7	543.0	543.1	648.9	562.1
34.7	Little Rock, Arkansas	50°	578.4	470.0	400.0	383.6	408.1	486.5	435.4
39.7	Indianapolis, Indiana	55°	501.7	354.3	293.2	300.1	332.1	417.6	360.4

Table 5-1: Btu Output per panel per month - one way to Simplify Sizing

Another example of a simplified procedure is shown in Table 5-2. This table gives a simple divisor termed "LC" for 85 cities that can be divided into a building's heat loss per degree day to arrive at a collector size to supply a specific fraction of the heating load. This is based on simulation studies made at Los Alamos Scientific Laboratory. Here is how it works.

Suppose a house located in Atlanta, Georgia is to be solar heated. Assume the heat loss is 40,000 Btuh for 20° F average outside and 70° F average inside temperature. Table 5-2 illustrates how the calculation is made.

City, State	Latitude (°N)	Elevation (ft)	Degree-days	LC (Btu/degree-day-ft°) where solar provides 25%, 50%, 75% of total heat		
				25%	50%	75%
Los Alamos, NM	36	7200	6600	107	41	21
Columbus, OH	40	760	5211	77	29	13
Corvallis, OR	45	236	4726	120	42	18
Davis, CA	39	50	2502	198	72	33
East Lansing, MI	43	878	6909	76	28	13
East Wareham, MA	42	50	5891	97	37	18
El Centro, CA	33	12	1458	547	206	97
Flaming Gorge, UT	41	6273	6929	111	43	21
Granby, CO	40	8340	5524	119	47	23
Toronto, Canada	44	443	6827	72	27	13
Griffin, GA	33	1001	2136	217	84	42
Winnipeg, Canada	50	820	10629	63	23	11
Ithaca, NY	42	951	6914	68	24	11
Inyokern, CA	36	2186	3528	232	88	42
ANL, Lemont, IL	42	750	6155	79	30	14
Newport, RI	41	50	5804	97	37	18
Laramie, WY	41	7240	7381	106	42	21
Page, AZ	37	4280	6632	128	48	23
Prosser, WA	46	840	4805	117	41	18
Pullman, WA	47	2583	5542	100	36	16
Put-In-Bay, OH	42	580	5796	68	24	11
Richland, WA	47	731	5941	100	35	15
Raleigh, NC	36	440	3393	133	52	25
Riverside, CA	34	1050	1803	391	152	74
Seattle, WA	48	110	4785	94	33	13
Sayville, NY	41	56	4811	98	38	18
Schenectady, NY	43	490	6650	63	24	11
Seabrock, NJ	39	110	4812	97	37	18
Shreveport, LA	32	220	2184	179	70	35
State College, PA	41	1230	5934	78	29	14
Stillwater, OK	36	910	3725	132	52	25
Tallahassee, FL	30	64	1485	288	113	57
Tucson, AZ	32	2440	1880	301	118	59
Oak Ridge, TN	36	940	3817	111	42	20
Fort Worth, TX	33	574	2405	185	73	37
Lake Charles, LA	30	60	1459	244	96	48
Apalachiola, FL	30	46	1308	324	129	65
Brownsville, TX	26	48	600	517	218	110
San Antonio, TX	30	818	1546	262	103	52
Greensboro, NC	36	914	3805	128	50	24
Hatteras, NC	35	27	2612	204	79	39
Atlanta, GA	34	1018	2961	154	59	29
Charleston, SC	33	69	2033	210	82	41
Nashville, TN	36	614	3578	117	44	21
Lake Charles, LA	30	39	1459	261	104	53
Little Rock, AR	35	276	3219	126	48	24
Oklahoma City, OK	35	1317	3725	134	53	26
Columbia, MO	39	814	5046	102	38	18
Dodge City, KA	38	2625	4986	126	49	24
Caribou, ME	47	640	9767	68	26	12
Burlington, VT	44	385	8269	63	24	11
Blue Hill, MA	42	670	6368	82	31	15
Cleveland, OH	41	871	6351	71	26	12
Madison, WI	43	889	7863	76	28	13
Sault Ste. Marie, MI	46	724	9048	74	27	12
Saint Cloud, MN	46	1062	8879	71	27	13
Lincoln, NE	41	1316	5864	104	39	19
Midland, TX	32	2885	2591	202	79	39
El Paso, TX	32	3954	2700	228	88	44
Albuquerque, NM	35	5327	4348	161	64	31
Grand Junction, CO	39	4832	5641	119	46	22
Ely, NV	39	6279	7733	119	47	23
Las Vegas, NV	36	2188	2709	218	84	42
Phoenix, AZ	33	1139	1765	300	118	59
Reno, NV	39	4400	6632	125	47	22
Santa Maria, CA	35	289	2967	353	142	67
Bismark, ND	47	1677	8851	78	29	14
Lander, WY	43	5574	7870	108	42	21
Glasgow, MT	48	2109	2996	105	41	20
Rapid City, SD	44	3180	7345	97	37	18
Salt Lake City, UT	41	4238	6052	107	40	19
Boise, ID	44	2895	5809	108	39	17
Great Falls, MT	47	3692	7750	93	35	16
Spokane, WA	48	2356	6655	90	31	14
Medford, OR	42	1321	5008	107	38	16
Los Angeles, CA	34	540	2061	416	157	75
Fresno, CA	37	336	2492	195	70	32
Silver Hill, MD	39	292	4224	111	43	21
Cape Hatteras, NC	35	27	4612	189	74	36
Sterling, VA	39	276	4224	111	43	21
Indianapolis, IN	40	819	5699	86	32	15
Astoria, OR	46	22	5186	127	45	19
Boston, MA	42	157	5624	86	33	16
New York, NY	41	187	4871	88	34	16
North Omaha, NE	41	1323	6612	89	34	16

Table 5-2: "Divisors" to Determine Collector Area Required to Supply a Fixed Percentage of Energy.

A third example of a simplified procedure can be found in Figure 5-7, which shows a portion of this table.

In this table, a separate divisor is provided for air and liquid collector systems. Further, there are two choices of collector tilt and a selection for 30, 50 or 70 percent solar contribution. Also, in this case, the divisor is simply divided into the calculated design heat loss, which differs from the previous example. Here is how to use the table.

Consider the previous example city, Atlanta, Georgia, and the building with a 40,000 Btuh heat loss. For a liquid system, Figure 5-7 provides divisors of 316 to obtain a 30 percent solar contribution, 152 for 50 percent, and 84 for a 70 percent contribution for a liquid collector tilted at 53 degrees. Based on a 50 percent contribution:

$$\frac{40,000}{152} = 263 \text{ square feet of collector.}$$

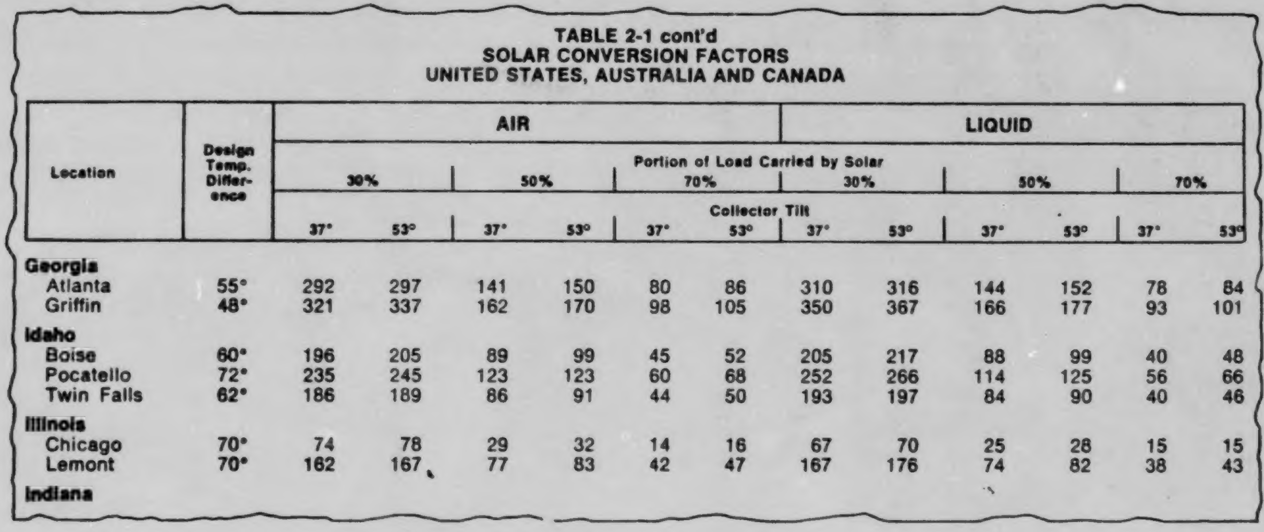


TABLE 2-1 cont'd
SOLAR CONVERSION FACTORS
UNITED STATES, AUSTRALIA AND CANADA

Location	Design Temp. Difference	AIR						LIQUID					
		Portion of Load Carried by Solar											
		30%		50%		70%		30%		50%		70%	
		Collector Tilt											
		37°	53°	37°	53°	37°	53°	37°	53°	37°	53°	37°	53°
Georgia													
Atlanta	55°	292	297	141	150	80	86	310	316	144	152	78	84
Griffin	48°	321	337	162	170	98	105	350	367	166	177	93	101
Idaho													
Boise	60°	196	205	89	99	45	52	205	217	88	99	40	48
Pocatello	72°	235	245	123	123	60	68	252	266	114	125	56	66
Twin Falls	62°	186	189	86	91	44	50	193	197	84	90	40	46
Illinois													
Chicago	70°	74	78	29	32	14	16	67	70	25	28	15	15
Lemont	70°	162	167	77	83	42	47	167	176	74	82	38	43
Indiana													

Figure 5-7: Portion of Table 2-1 SMACNA Installation Standards

Note, first, that the estimated collector area determined from the procedures used in Figure 5-8 and above differ substantially (325 vs. 263). Second, the Los Alamos procedure made no distinction between air or liquid systems. Also, the designer had a choice of tilt angles in this procedure, but he did not in the Los Alamos approach. (Los Alamos assumed a tilt angle of latitude plus 10 degrees which, for Atlanta, would mean a collector tilt of 44 degrees).

The important point in this comparison is that "assumed conditions" for specific simplified procedures are not always the same. Before a designer/technician uses any simplified approach for collector sizing, there should be a thorough understanding of the assumptions made by the developers of the design technique. The designer/technician should also be aware of how these assumed parameters differ for specific applications.

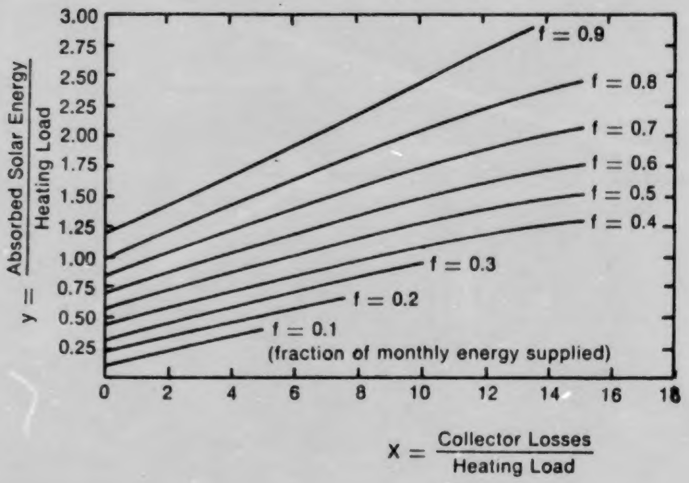


Figure 5-8: f-Chart for liquid-based Solar Heating Systems.

F-CHART Collector Selection Technique

Perhaps the best known detailed sizing procedure and the one most frequently referred to for specific collectors is termed the FCHART procedure, developed at the University of Wisconsin. (Figure 5-7 is based on this procedure). Figures 5-8 and 5-9 show the FCHART for liquid systems and air systems respectively. Each "f" curve represents the fraction of the monthly energy demand supplied by solar energy as a function of the "X" and "Y" coordinates. The horizontal scale (X) is the ratio of the monthly solar collector losses to the monthly heating demand; the "Y" scale is the ratio of the monthly solar energy absorbed by the collector to the monthly heating demand. Since values of X and Y vary each month, the fraction (f) of the heating demand satisfied by the collector will also vary each month.

The curves were derived from computer simulation studies of two "standard" air and liquid solar heating systems. To use the FCHART, the slope and intercept of the efficiency curve for the specific manufacturer's collector being used must be known. In addition, the designer must know the monthly incident solar

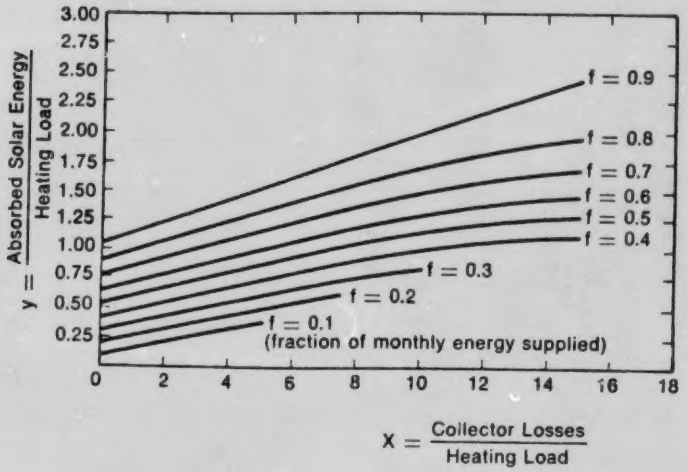


Figure 5-9: f-Chart for Solar Air Heating Systems

radiation, monthly degrees days, building heat loss, and average monthly outdoor temperature. There are also "correction factors" to be applied for "non-standard" variations in system configuration, such as larger or smaller storage capacity than "standard," and heat exchanger performance for liquid systems.

Values of X and Y are calculated for each month of the year for the locality in question and for an assumed collector area. Monthly values of "f" are obtained from the chart at the intersection of the X and Y coordinates. The monthly solar energy contributions are then totalled for the season and divided by the total heating demand to determine a seasonal value of "f". The procedure is then repeated for several other assumed collector areas.

The results could be as follows for a specific building and collector type: collector area (assumed) of 400 square feet - seasonal "f" equals .43; collector area increased to 600 square feet - "f" increases to .55; and for a collector area of 800 square feet - "f" equals .64. Thus, to supply just over half the heating needs (.55), a collector of 600 square feet would be required in this hypothetical case.

The University of Wisconsin has developed a computer program utilizing the FCHART procedure. It is available for sale for private use and a number of collector manufacturers make this service available to their customers. Details of the manual procedure to use FCHART are included in HUD's Intermediate Minimum Property Standards Supplement.

Between the highly sophisticated FCHART computer analysis and the drudgery of manual calculations, there is a hand calculator approach devised by researchers at Colorado State University. Termed "Relative Areas Analysis" a designer equipped with special tabulated date for specific cities can simply "plug in" a few numbers into a calculator and determine an annual load fraction. Details of the procedure are contained in a thesis written by C. Dennis Barley, Department of Mechanical Engineering, CSU, Fort Collins, CO 80523. The procedure also involves an economic analysis to determine the "best size" of collector. The concept of life-cycle-costing will be discussed next.

ECONOMICAL COLLECTOR SIZING

There are many reasons why a customer/client may choose to purchase a solar heating system. Among them is a concern for the environment, fear of fossil fuel shortages, and a desire to have a new and innovative heating system. However, when the time comes to pay for the heating system, nearly all customers/clients will want to know how much the solar system will save before deciding to purchase. In terms of economic benefits, what size system might be best for a customer; one that contributes 20, 30, 50, or 80 percent of the energy need? Figure 5-10 illustrates a typical economical analysis. The top curve relates collector area versus fraction of the load supplied, as might be determined from F-Chart, the LC table or other simplified procedures.

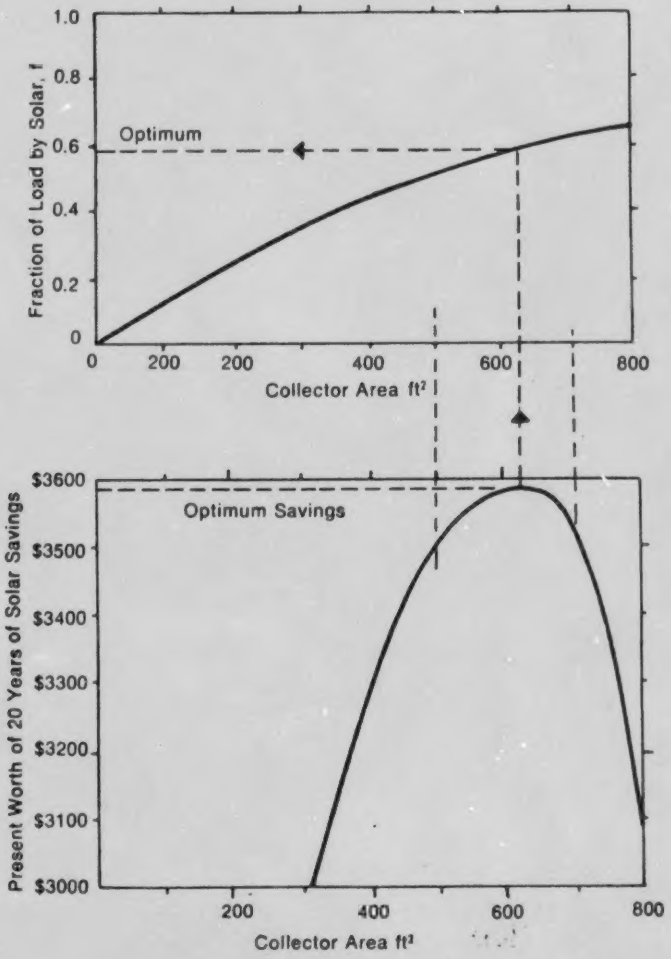


Fig. 5-10: Savings versus collector area.

The lower curve (fig. 5-10), relates collector area to money saved over a 20-year period. In this example, the peak savings are realized for a collector area of 630 square feet and would result in a 60% solar contribution. Please note, however, that a modest change of $100 out of the projected $3600 savings in this example would alter the desirable collector size from a low of 500 square feet to perhaps 700 square feet. Thus, the optimum plateau is fairly flat and this means one's choice is rather broad in terms of optimizing payback, especially because of the many assumptions made to complete an economic analysis. Presently, it appears that collectors selected to serve from 50 to 70% of the heating load are most economic in normal installations.

LIFE-CYCLE COSTS

Because of the need to install a combination system (solar plus conventional heating equipment), it is obvious that it will be impossible to create a solar assisted heating system which is less expensive than a conventional system based on initial costs. The sale will be made based on what is termed "life-cycle cost" (see fig. 5-11 on the following page). The customer/client must be convinced that the savings in energy cost over the years the system will actually last (before it wears out), will offset the initial cost of installing the solar heating system. In making the determination of life-cycle costs, it is necessary to consider the following factors:

1. Solar System Fixed Initial Cost.
2. Solar Collector installed cost per square foot.
3. Loan interest rate.
4. Loan term.
5. Loan down payment.
6. Property tax rate.
7. Income tax rate.
8. Maintenance costs.
9. Insurance.
10. Property taxes.
11. Present fuel costs.
12. Inflation

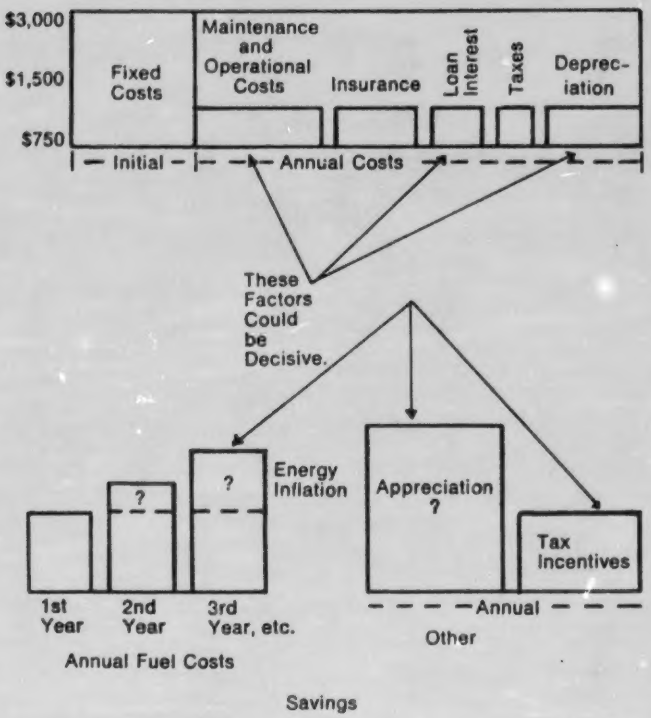

$3,000
$1,500
$750
Fixed Costs
Maintenance and Operational Costs
Insurance
Loan Interest
Taxes
Deprec-iation
Initial
Annual Costs
These Factors Could be Decisive.
?
?
Energy Inflation
Appreciation ?
Tax Incentives
1st Year
2nd Year
3rd Year, etc.
Annual Fuel Costs
Annual
Other
Savings

Figure 5-11: Life cycle costing is a method whereby the total costs of a product can be measured against the annual savings, showing the buyer approximately when his or her investment is paid for.

Obviously, these factors will vary according to the location of the installation. It will be necessary to become familiar with the values for each of the factors for a specific location. The savings versus collector area as shown in Figure 5-10 involves a great deal of manual calculation. Building owner cash flows are calculated for each year of the analysis for both solar and non-solar space heating installations. By comparing the present values of the yearly costs of the solar and non-solar systems, the economic feasibility of the solar system is determined. Present value refers to the savings in terms of today's dollars. As one knows, the value of money is time-related, and is a normal factor to consider when analyzing investments.

"SOLCOST" COMPUTER ANALYSIS

Because of the numerous calculations involved, the computer can be put to ideal use. An ARDA funded program called "Solcost" is available to contractors to determine optimum collector size. The cost is from $10 to $20 for sizing and life-cycle-costing. For an additional $40, a heat loss calculation can be obtined by computer. Figure 5-12, shown on the following page, illustrates the type of Solcost residential analysis that would be provided.

For further information on the Solcost procedure contact: International Business Services, Solar Group, 1010 Vermont Avenue, Washington, D. C. 20005, telephone, (202) 628-1450.

EFFECT OF MOUNTING SOLAR COLLECTORS AT ANGLES OTHER THAN OPTIMUM

For space heating (only) applications, the optimum angle of a solar collector with respect to the horizon if 15° plus the local latitude. For heating-cooling applications (where collector heat is also used for cooling), latitude plus 5°; for domestic water heating only, tilt should be equal to latitude. For Columbus, Ohio, the latitude is 40° N; therefore, the optimum tilt for heating is $40 + 15 = 55^{\circ}$ from horizontal. For some installations, it may be impractical to maintain an optimum tilt. Before deciding to install collectors at the optimum angle, it is very important that the effect on the efficiency of the collector be determined, and then consider the cost of the additional collector area needed versus the expense of an elaborate frame.

Figure 5-13, shown in the following pages, graphically illustrates the effect of changing the tilt on the efficiency of the collector. The computation of the additional collector area required is also included in Figure 5-13.

EFFECT OF FACING THE COLLECTOR EAST OR WEST OF DUE SOUTH

When a collector must be oriented east or west of due south, more of the sun's energy which strikes it is lost. As in the case of changing the tilt, this required that the collector area be increased to compensate for this loss. The graph in Figure 5-14, shown in the following pages, will enable you to determine the efficiency of a collector which is oriented east or west of due

INPUT

Input Parameter	User Input
Solar System Type	1
Fuel Type for Reference Heating System	2
Fuel Type for Solar Auxiliary Heating System	2
Collector Type	3
Collector Tilt Angle	55. (Degrees)
Collector Azimuth Angle	+ 10. (Degrees)
Site Location	DENVER
Building Heat Loss Coefficient	8.3 (BTU/Sq. Ft./Deg.-Day)
Building Floor Area	1950. (Sq. Feet)
Solar System Fixed Initial Cost	$1000.
Solar Collector Installed Cost/Sq. Ft.	$12.00
Loan Interest Rate	.09 (9 percent)
Loan Term	20. (Year)
Loan Down Payment	.22 (22 percent)
Property Tax Rate	.02 (2 percent)
Income Tax Rate	.30 (30 percent)
Inflation of Maint., Insur. Property Taxes	.04 (4 percent)
Present Electricity Cost $/Kw-hr	$.035
Electricity Cost Escalation Per Year	.10 (10 percent)

EXPLANATION OF SELECTED INPUT VALUES

Solar System Type
This input parameter covers different types of solar systems used for heating and cooling of buildings. For example, the indicator (1) above signifies space heating with liquid collectors, collector/storage heat exchanger, fan coils or air duct heat exchanger systems.

Fuel Type for Reference (Conventional) Heating System
Fuel types include natural gas, electricity, fuel oil, LP gas and coal. When you input an indicator (2) as above, it means electricity is the fuel used for the reference or conventional heating system.

Fuel Type for Solar Auxiliary Heating System
These fuel types are usually the same as those for the reference heating system input parameter — natural gas, electricity, fuel oil, LP gas and coal. The indicator (2) represents electricity.

Collector Type
All collector types including liquid, air, evacuated tube, and others can be defined by this parameter. The indicator (3) represents a liquid, flat plate, 1 cover, selective absorber collector.

OUTPUT

COLLECTOR SIZE OPTIMIZATION BY SOLCOST

Collector type = flat plate 1 glass selective
Best solar collector size for tilt angle of 55 degrees is 400 sq. ft.
Solar costs = 1000 fixed + 4800 collector + 900 storage

Input conventional system costs = 0
Initial solar investment = $6700 Down payment = $1500
Financial scenario — residence

CASH FLOW SUMMARY

Yr.	(A) Fuel/Utility Savings	(B) Maint. + Insur.	(C) Property Tax	(D) Annual Interest	(E) Tax Savings	(F) Loan Payment	(G) Net Cash Flow
							-1500 (Down Payment)
1	500	70	135	468	181	570	-94
2	550	73	140	459	180	570	-53
3	605	76	146	449	178	570	-8
4	665	79	152	438	177	570	42
5	732	82	158	426	175	570	98
6	805	85	164	413	173	570	159
7	886	89	171	399	171	570	228
8	974	92	178	384	168	570	303
9	1072	96	185	367	166	570	387
10	1179	100	192	349	162	570	480
11	1297	104	200	329	159	570	582
12	1427	108	208	307	155	570	696
13	1569	112	216	284	150	570	821
14	1726	117	225	258	145	570	960
15	1899	121	234	230	139	570	1113
16	2089	126	243	199	133	570	1283
17	2297	131	253	166	126	570	1470
18	2527	136	263	130	118	570	1676
19	2780	142	273	90	109	570	1904
20	3058	147	284	47	99	570	2156
Totals	28637	2086	4020	6192	3064	11400	12703

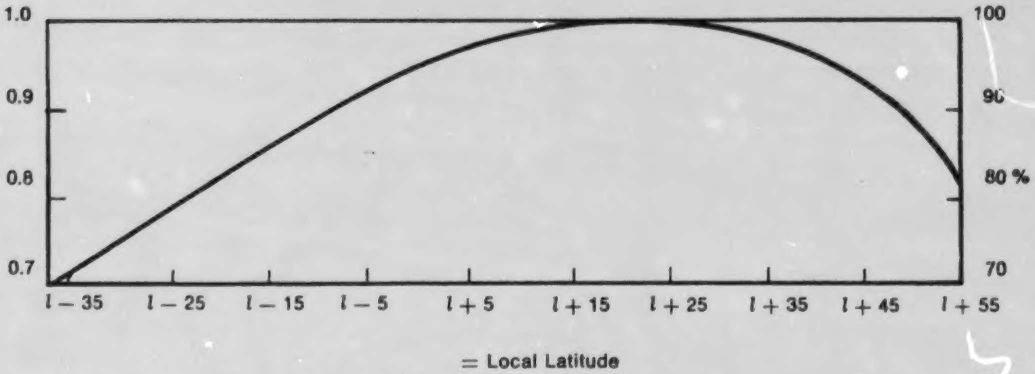
Payback time for net cash flow to equal down payment	8.9 years
Payback time for net cash flow to equal down payment	9.9 years
Rate of return on net cash flow	16.3 percent
Annual portion of load provided by solar	72.0 percent
Annual energy savings with solar system	91.3 million btus

Tax savings = income tax rate × (C + D)
Net cash flow = A − B − C + E − F

* Similar calculations can be made for businesses and non-profit organizations where special considerations such as depreciation and tax deductions are accounted for.

Figure 5-12: An Example of SOLCOST Use for Residential Homeowner*.

south. For example, a collector oriented 45° east of south will be 90% efficient. Using the same formula as given in Figure 5-13, it can be seen that the collector must again be increased by 11% to offset the loss due to the orientation.



Example of How to Compute Increased
Collector Area Required

1. A collector in Columbus, Ohio, is located at a tilt of 25°.
2. Optimum for Columbus, Ohio, is 40+ 15 =55.
   Therefore, the collector tilted at an angle of $l - 15$.
3. From this angle the collector will operate at 90% efficiency.
4. To compute the increased collector area required (a) use the following formula.

$$a := \frac{1}{e} - 1$$

Where
e = efficiency of the collector (decimal)

$$a = \frac{1}{.09} - 1 = 11\%$$

Therefore, the area of the collector must be increased by 11%.

Figure 5-13: Effect of solar collector tilt on solar heating performance.

As mentioned earlier, there are many methods for sizing the array; however, this module will concentrate efforts on one method. The method chosen for proficiency is a manufacturer's worksheet method that incorporates the f-Chart.

The following is an example calculation, including instructions, of the worksheet method used by Lennox Corporation. Assume the hot water and space heated load to be as stated in the example.

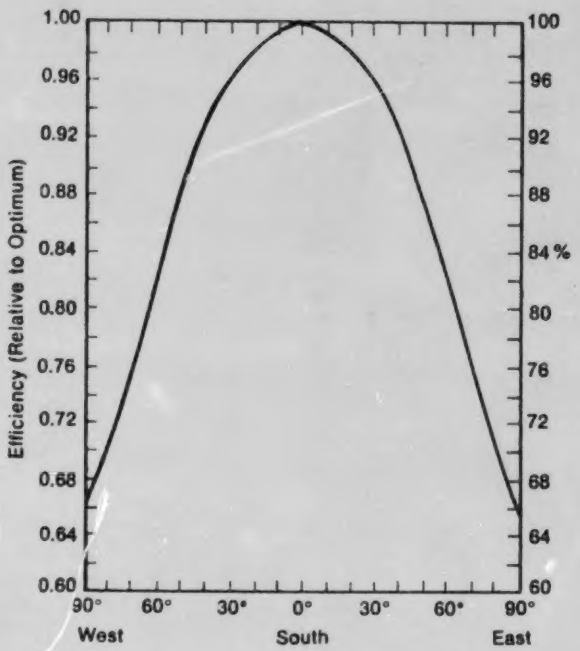


Figure 5-14: Effect of Solar Collector Orientation on Solar Heating Performance.

EQUIPMENT SELECTION

Selecting properly sized equipment is critical to the operating success of any solar system. Detailed, step-by-step worksheets have been created to make the sizing process easier. There are five main steps in sizing a complete solar system. They are:

1. Collector sizing - short form.

2. Collector sizing - long form.

3. Heat Exchanger, Purge Coil, Storage Tank and Solar Space Heat Coil Sizing.

4. Hydronic Components Sizing, and

5. Auxiliary Furnace and Hot Water Heater Sizing.

The collectors are selected first because they are the heart of the solar system, and are usually the most expensive of the system components. Care must be taken when choosing the number of collectors, for this decision will greatly affect economic payback and ultimately customer satisfaction. The number of collectors is dependent upon four main factors:

1. total heat load;
2. percentage of the total load to be carried by solar energy;
3. the amount of money the buyer is willing to spend on the system; and
4. the amount of suitable space (for example, south-facing roof) available on which to mount the collectors.

Short form method and long form method worksheets have been developed to facilitate collector sizing. The short form (Step 1) provides an estimation of collector area, enabling the designer quickly to qualify the buyer in terms of money and space. The long form (Step 2) provides more accurate documentation of collector performance.

The heat exchanger, purge coil, storage tank and solar space heat coil (Step 3) are selected, based on the size (square ft.) of the collector array. Hydronic components (i.e., piping, pumps, valves) are selected in Step 4, based on collector array size, flow rate, pressure drop and other hydronic principles. Step 5 sizes the auxiliary furnace and water heater.

To aid the student's understanding, sample worksheets are filled out for an example solar system home. The solar system is the type previously described, with space heat and hot water capabilities. The example home is in Ames, Iowa. The house is a newly constructed, three bedroom, ranch style home (1400 sq. ft.). A family of four will occupy the home. (A heating and cooling load summary sheet for the example home is on the following page.) Detailed worksheet instructions succeed some of the worksheets. An extensive data bank in the next section has information to complete the worksheets.

V-S-132

FORM J—1

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Contractors Association
1501 Wilson Boulevard
Arlington, Virginia 22209
Printed in U.S.A.

January, 1968

Plan No.
Date
Calculated by

WORKSHEET FOR MANUAL J
LOAD CALCULATIONS FOR RESIDENTIAL AIR CONDITIONING

For: Name JOHN DOE
Address 1504 SECOND AVE.
City and State or Province AMES, IOWA
By: Contractor ACME HEATING AND AIR CONDITIONING
Address 312 MAIN
City AMES, IOWA

Winter Design Conditions

Outside -10° F Inside 70° F Temperature Difference 80 Degrees

(Insert data below only after all heat loss calculations have been completed)

Total Heat Loss (Btuh) 55,440 (From Line No. 15) Model No. E12Q3-20
Serial No. 5875G00015 Manufactured by LENNOX
Rating Data: Input 82000 Btuh Output at Bonnet 65600 Btuh
675 Cfm
Description of Controls T872C HONEYWELL THERMOSTAT
Q672B1004 SUBBASE

Summer Design Conditions

Outside 90° F Inside 75° F
North Latitude 40° Degrees Daily Range M

(Insert data below only after all heat gain calculations have been completed)

Total Heat Gain (Btuh) 19,820 (From Line No. 20 or 21, if used)
Equipment Capacity Multiplier 1.00 Model No. HS9-211
Serial No. 5175J15104 Manufactured by LENNOX
Rating Data: Cooling Capacity 19,950 Btuh Air Volume 675 Cfm
Description of Controls
C4-26FF S/T = 0.71
L10-26-25

Winter Construction Data (See Table 2)

Walls and Partitions WOOD FRAME CONSTRUCTION 3" INSULATION (R-11) BASEMENT - CONCRETE 3' ABOVE AND 5' BELOW
Windows and Doors PICTURE WINDOW - THERMOPANE DOUBLE HUNG WINDOWS - WITH STORM SASH BASEMENT WINDOW - SINGLE GLASS DOORS WEATHER STRIPPED WITH STORM
Ceilings VENTED ATTIC WITH 12" INSULATION (R-30)
Floors BASEMENT - CONCRETE

Summer Construction Data (See Table 5)

Direction House Faces SOUTH
Windows and Doors DRAPERIES
Walls and Partitions
Ceilings DARK ROOF
Floors

SAME AS WINTER

FILE

COLLECTOR SIZING - SHORT FORM METHOD

INTRODUCTION

The short form method for collector sizing is the first step in the system designing process. Step 1A Worksheet employs an LC (Load Collector) factor as the basis for estimating collector area. By using this short form, the designer can quickly estimate the collector area needed to achieve a certain solar percentage (25%, 50%, and 75%) of the total space heat and domestic hot water load.

Approximate cost of the solar system can also be figured from the collector area estimate. This means that within minutes, the designer can qualify the prospective buyer with respect to the prime factors of money and space.

NOTES: The short form method for collector sizing is only an estimation based on a number of assumptions, and should not be used in lieu of the long form method. Another short form means of estimating collector area employs an SA/L factor. However, Step 2 (collector sizing - long form) must be completed prior to initially using the SA/L method for a given area.

V-S-134

STEP 1A. COLLECTOR SIZING — SHORT FORM METHOD WORKSHEET

Line 1. Design Heat Loss of Structure (Btuh): (from standard heat loss calculation) .... 55,440 Btuh

Line 2. Winter Design Temperature Difference (°F): (Indoor Design Temp. 70) – (Outdoor Design Temp. -10) = 80 °F

Line 3. Space Heat Load (Btu per degree-day):

$$\frac{(\text{Line 1 } 55{,}440) \times 24 \times .75}{(\text{Line 2 } 80)} = \ldots\ldots\ldots\ldots\ldots\ldots\ 12{,}474 \text{ Btu/D-day}$$

Line 4. Desired Annual Solar Percentage of Total Load: 25%, 50% or 75% .... 50 %

Line 5. Approximate Total Collector Area (sq. ft.):

$$\frac{(\text{Line 3 } 12{,}474)}{\text{LC Factor } 46 \text{ (from LC Factor Table for \_\_\_\_\_)}} = \ldots\ldots\ldots\ldots\ 271 \text{ sq. ft.}$$

Line 6. Effective Absorber Area Per Collector (from Engineering Handbook Solar Collector sheet) .... 15.4 sq. ft.

Line 7. Estimated Number of Collectors:

$$\frac{(\text{Line 5 } 271)}{(\text{Line 6 } 15.4)} = \ldots\ldots\ldots\ldots\ 17.6 \text{ OR } 18 \text{ COLLECTORS}$$

Line 8. Effective Area of Collector Array (sq. ft.): (Line 6 15.4) x (Line 7 18) = ... 277 sq. ft.

LC FACTOR TABLE

CITY, STATE	LATITUDE	SOLAR PERCENTAGE OF TOTAL LOAD		
		25%	50%	75%
Ames, Ia.	42.0	134	46	19
Albuquerque, N.M.	35.0	334	120	60
Atlanta, Ga.	33.4	316	109	46
Boulder, Colo.	40.0	191	74	35
Columbus, Oh.	40.0	131	40	14
Dallas, Tx.	32.5	416	133	58
Davis, Ca.	38.3	394	120	46
Miami, Fl.	25.5	1443	646	382
Norfolk, Va.	36.5	270	90	40
San Diego, Ca.	32.4	459	211	112
Edmonton, At.	53.3	96	35	14
Moncton, N.B.	40.0	84	25	8
Toronto, Ot.	43.4	98	30	10
Vancouver, B.C.	48.6	102	28	8
Winnipeg, Ma.	49.5	94	33	13

INSTRUCTIONS FOR STEP 1A WORKSHEET

Line 1. Industry-approved procedures should be followed to calculate design heat loss, such as outlined in *NESCA* Load Calculation Manual J* or *ASHRAE* Fundamentals Guide*. (See sample summary sheet of *NESCA Manual J Worksheet* on page 5 of this section.)

NOTE: Heat loss calculation procedures are the same for a "solar" residence as a "non-solar" residence.

Line 2. Winter Design Temp. Difference = Indoor Design Temp. – Outdoor Design Temp. **NOTE:** Outdoor winter design temp. (°F) is listed on the Weather and Radiation Tables for 15 sites in the DATA portion of this section. Indoor design temp. is determined by the system designer and/or homeowner.

Line 3. Calculate space heat load in Btu per degree-day:

$$\frac{(\text{Line 1}) \times \overset{\textcircled{a}}{24} \times \overset{\textcircled{b}}{.75}}{(\text{Line 2})}$$

ⓐ 24 stands for number of hours per day. This step converts heat loss from Btu per hour to Btu per day.

ⓑ .75 is a proportionality factor. This factor modifies heat loss calculations to take into consideration current construction practices (i.e. increased insulation) which reduce heat loss.

NOTE: Degree-day is a unit of measurement for determining energy use. For further information see Step 2A Worksheet Instructions.

Line 4. Select one of the following solar percentages of total load:

25%, 50% or 75%

NOTE: The designer and/or homeowner must choose one of the above solar load percentages in order to select the proper LC factor in Line 5. The optimum solar load percentage in terms of economic payback is usually between 50% and 75%.

Line 5. Calculate approximate total collector area for desired solar load percentage:

$$\frac{(\text{Line 3})}{\text{LC factor (from LC Factor Table)}}$$

NOTE: Find appropriate LC (Load Collector) factor from LC Table on worksheet. LC factor is ratio between the building load and the collector area needed to obtain a certain solar percentage of that load.

Line 6. Effective absorber area per Lennox collector can be found in Engineering Handbook Collector sheet. (This sheet is in DATA portion of this section.) Effective area of both the Lennox LSC18-1 double glass collector and the Lennox LSC18-1S single glass collector is 15.4 sq. ft.

Line 7. Calculate estimated number of collectors:

$$\frac{(\text{Line 5})}{(\text{Line 6})}$$

Line 8. Calculate effective absorber area of collector array (sq. ft.):

(Line 6) x (Line 7)

**NESCA stands for National Environmental Systems Contractors Association. ASHRAE stands for American Society of Heating, Refrigeration and Air Conditioning Engineers.*

COLLECTOR SIZING - LONG FORM METHOD

INTRODUCTION

The long form method for collector sizing is used after the designer has qualified the buyer with the short form. The long form provides more accurate collector performance documentation than the short form. Such documentation is normally required by banks, lending institutions or funding agencies (i.e. HUD, FHA).

Included in the long form are monthly and yearly breakdowns of heat load and solar percentages of the total heat load. These breakdowns are based on several estimated collector areas. The designer and buyer are able to compare the merits of the various collector areas and choose that area which best suits their needs.

Three worksheets comprise the long form:

Step 2A: Worksheet figures monthly and yearly heat load for space heat and domestic hot water;

Step 2B: Worksheet figures monthly and yearly solar load percentages, based on collector area estimate from short form.

Step 2C: Worksheet is an abbreviated version of Step 2B Worksheet. If the designer desires additional monthly and yearly solar load percentage input for other collector areas, the worksheet provides a quick way to figure this.

COLLECTOR SIZING — LONG FORM METHOD

STEP 2A. HEAT LOAD WORKSHEET

(Instructions for this worksheet on next page.)

Line 1. Design Heat Loss of Structure (Btuh) .... 55,440 Btuh
(from standard heat loss calculation)

Line 2. Winter Design Temperature Difference (°F):
(Indoor Design Temp. 70) - (Outdoor Design Temp. -10) = .. 80 °F

Line 3. Space Heat Load (Btu per degree-day):
(Line 1 55,440) x 24 x .75) / (Line 2 80) = ... 12,474 Btu/D-day

Line 4. Hot Water Temperature Difference (°F):
(Setpoint Temperature 140°F) –
(Cold Water Supply Temperature 55°F) = ... 85 °F

Line 5. Hot Water Consumption (gal. per day) .... 80 gal./day

Line 6. Hot Water Load (Btu per day):
(Line 4 85) x (Line 5 80) x 8.33 = ... 56,644 Btu/day

Column A		Column B	Column C	Column D	Column E
DAYS PER MONTH		HEATING DEGREE-DAYS (from Wea. & Rad.Tables for____)	SPACE HEAT LOAD (Btu/mo.): (Line 3) x (Column B) x (Abbreviation Factor*)	WATER HEAT LOAD (Btu/mo.): (Line 6) x (Column A) x (Abbreviation Factor*)	TOTAL HEAT LOAD (Btu/mo.): (Column C Abbreviation) + (Column D Abbreviation)
Jan.	31	1429	17.83 x 10^6	1.76 x 10^6	19.59 x 10^6
Feb.	28	1151	14.36 " "	1.59 " "	15.95 " "
Mar.	31	970	12.10 " "	1.76 " "	13.86 " "
Apr.	30	468	5.84 " "	1.70 " "	7.54 " "
May	31	191	2.38 " "	1.76 " "	4.14 " "
June	30	32	0.40 " "	1.70 " "	2.10 " "
July	31	0	0.00	1.76 " "	1.76 " "
Aug.	31	15	0.19 " "	1.76 " "	1.95 " "
Sept.	30	105	1.31 " "	1.70 " "	3.01 " "
Oct.	31	370	4.62 " "	1.76 " "	6.38 " "
Nov.	30	834	10.40 " "	1.70 " "	12.10 " "
Dec.	31	1259	15.70 " "	1.76 " "	17.46 " "
TOTAL	365	6824	85.13 " "	20.71 " "	105.84 " "

**The abbreviation factor abbreviates and rounds a large number by moving decimal six digits to the left; rounding to two decimal numbers; and multiplying by 10^6. (See Step 2A Worksheet Instructions for further explanation.)*

INSTRUCTIONS FOR STEP 2A WORKSHEET

Lines 1-3 These lines are identical to lines 1-3 of STEP 1A Worksheet. See Step 1A Worksheet Instruction for explanation.

Line 4. Determine hot water temperature difference:

ⓐ
(Water Heater Setpoint Temperature) —
ⓑ
(Cold Water Supply Temperature)

ⓐ Water heater setpoint temperature is selected by homeowner; 120°-140°F is normal setpoint.

ⓑ Cold water supply temperature is temperature of water entering house; consult local water department or take temperature reading from cold water tap.

Line 5. Determine daily hot water consumption. If exact consumption is not known, a rule of thumb is 20 gallons per each family member.

Line 6. Calculate hot water load in Btu per day:

(Line 4) x (Line 5) x 8.33

NOTE: 8.33 converts gallons to pounds so that heat load in Btu can be figured. 8.33 lb. = 1 gal. of water.

Col. B. Find heating degree-days for solar home locale from WEATHER AND RADIATION TABLES. Tables for 15 sites are in DATA portion of this section.

NOTE: A degree-day is a unit of measure used in determining energy usage for space heating. A degree-day for heating is based on two assumptions: 1) Over a long period of time, passive solar and internal heat gains will offset the heat loss of a residence when the mean daily outdoor temperature is 65°F; and 2) The heat load will be proportional to the difference between 65°F and the mean daily temperature.

Degree-days for heating are figured by subtracting mean daily temperature from 65°F. If the mean temperature for a certain day is 55°F, there are 10 degree-days for that day (65°-55° = 10 degree-days). If the mean temperature for that day is 65°F or above there are 0 degree-days.

Col. C, D, E. Complete the table by filling in these columns as directed on worksheet.

NOTE: Columns C, D and E contain an Abbreviation Factor. The Abbreviation Factor abbreviates and rounds a large number to a smaller, more workable number. This is done by moving the decimal six digits to the left, rounding to two decimal numbers and multiplying by 10^6 ($10^6 = 1,000,000$). **Example**: In Column C for the month of January, assume multiplying Line 3 by Column B yields 19,014,274. Using the Abbreviation Factor 19,014,274 becomes 19.01×10^6.

COLLECTOR SIZING — LONG FORM METHOD

STEP 2B. SOLAR LOAD PERCENTAGE OF TOTAL LOAD WORKSHEET

(Instructions for this worksheet on next page.)

Line 7. Estimated Effective Absorber Area of Collector Array (from line 8 of Step 1A Worksheet or SA/L Short Form Worksheet .... 277 sq. ft.

Line 8. Collector Performance Curve Slope (from Engineering Handbook data) .... .60

Line 9. Collector Peformance Curve Intercept (from Engineering Handbook data) .... .78

Line 10. Collector to Storage Heat Exchanger Factor (.95 is factor for Lennox systems)........ .95

Line 11. Collector Orientation Factor (.95 is factor for Lennox systems)........ .95

Line 12. Preliminary "X" Factor of F-Chart:
(Line 7 277) x (Line 8 .60) x (Line 10 .95) = .. 157.89

Line 13. Preliminary "Y" Factor of F-Chart:
(Line 7 277) x (Line 9 .78) x (Line 10 .95) x (Line 11 .95) = ... 194.99

Col. F	Col. G	Col. H	Col. I	Col. J	Col. K	Col. L	Col. M	Col. N
DAYS PER MONTH	HOURS PER MONTH	AVERAGE AMBIENT AIR TEMP. (°F): (from Wea. & Rad. Tables)	TEMP. FACTOR: 212-(Col. H)	"X" COORDINATE: (Line 12) x (Col. G) x (Col. I) x (Abbreviation factor*) (Col. E, Step 2A Worksheet Abbreviation)	RADIATION ON TILTED SURFACE (from Wea. & Rad. Tables) Tilt 50° Azm. 0	"Y" COORDINATE: (Line 13) x (Col. F) x (Col. K) x (Abbreviation Factor*) (Col. E, Step 2A Worksheet Abbreviation)	SOLAR PERCENTAGE (from F-Chart)	SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. M)
J-31	744	19	193	1.16	1283	.40	.31	6.07×10^6
F-28	696	25	187	1.29	1509	.52	.39	6.22 " "
M-31	744	32	180	1.53	1506	.66	.49	6.79 " "
A-30	720	48	164	2.47	1469	1.14	.73	5.50 " "
M-31	744	59	153	4.34	1501	2.19	1.00	4.14 " "
J-30	720	68	144	7.80	1576	4.39	1.00	2.10 " "
J-31	744	73	139	9.28	1609	5.52	1.00	1.76 " "
A-31	744	72	140	8.43	1572	4.87	1.00	1.95 " "
S-30	720	63	149	5.44	1541	2.99	1.00	3.01 " "
O-31	744	52	160	2.94	1495	1.41	.85	5.42 " "
N-30	720	36	176	1.65	1290	.62	.48	5.81 " "
D-31	744	25	187	1.26	1095	.38	.29	5.06 " "
							YEARLY TOTAL	53.83 " "

Line 14. Yearly Solar Load Percentage:
(Col. N. Abbreviation TOTAL 53.83×10^6)
(Col. E, Step 2A Worksheet Abbreviation TOTAL 105.84×10^6) = ... 50.8 %

The Abbreviation Factor abbreviates and rounds a large number by: moving decimal six digits to the left; rounding to two decimal numbers; and multiplying by 10^6. (See Step 2A Worksheet Instruction for further explanation.)

INSTRUCTIONS FOR STEP 2B WORKSHEET

Line 7. Enter the approximate collector area desired as calculated in either Line 8 of the Step 1A Worksheet or the SA/L Short Form Worksheet.

Line 8. Determine collector performance curve slope from Lennox Engineering Handbook Solar Collector sheet in DATA portion of this section. (See following pages).

NOTE: Slope for Lennox LSC18-1 double glass collector is .60:

$$\frac{\text{"Y" axis intercept}}{\text{"X" axis intercept}} = \frac{.78}{1.3} = .60$$

Slope for Lennox LSC18-1S single glass collector is .81:

$$\frac{\text{"Y" axis intercept}}{\text{"X" axis intercept}} = \frac{.84}{1.04} = .81$$

Line 9. Find "Y" axis intercept of collector performance curve from Engineering Handbook Solar Collector sheet (in DATA portion of this section).

NOTE: Intercept of Lennox LSC18-1 collector performance curve is .78; LSC18-1S intercept is .84. (See "Slope and Intercept" portion of SOLAR ENERGY COLLECTION section for further explanation.)

Line 10. Find collector to storage heat exchanger factor.

NOTE: This factor for Lennox equipment is .95.

Line 11. Find collector orientation factor.

NOTE: This factor for Lennox Collectors is .95.

Line 12. Calculate preliminary "X" factor of F-Chart:

(Line 7) x (Line 8) x (Line 10)

NOTE: F-Charts are graphs used to determine solar percentage of total load and are located in the DATA portion (Tables 2a-2d) of this section.

The F-Charts are the result of correlating hundreds of detailed simulations of solar heating systems. For standard system configurations, the F-Charts eliminate the need for detailed simulations using hourly meteorological data.

Line 13. Calculate preliminary "Y" factor of F-Chart.

(Line 7) x (Line 9) x (Line 10) x (Line 11)

Col. H. Enter monthly average ambient temperatures for appropriate location from WEATHER AND RADIATION TABLES (tables for 15 sites are in DATA portion of this section).

Col. I. Calculate average ambient air temperature factor for each month:

212 — (Col. H)

NOTE: 212 is simply a reference number, and has no other significance.

Col. J. Calculate "X" coordinate of F-Chart for each month.:

$$\frac{\text{(Line 12) x (Col. G) x (Col. I) x (Abbrev. Fac.)}}{\text{(Col. E, Step 2A Worksheet Abbrev.)}}$$

NOTE: The Abbreviation Factor is the same as used in Step 2A Worksheet. (See Step 2A Worksheet Instructions for further explanation.)

NOTE: The 10^6 factor in numerator and denominator cancel, leaving a simple division calculation. **Example:** Assume for the month of January the numerator is figured to be 23.65×10^6 and the denominator 20.77×10^6. The two 10^6 factors cancel, and 23.65 divided by 20.77 yields 1.14.

$$\frac{23.65}{20.77} = 1.14$$

Col. K. Enter selected collector tilt and azimuth angles at top of graph, then enter the appropriate average daily radiation values for the selected location from WEATHER AND RADIATION TABLES (tables for 15 sites are in DATA portion of this section).

NOTE: Review THE SUN section for further explanation on how to select collector tilt and azimuth angles.

Col. L. Calculate "Y" coordinate of F-Chart for each month:

$$\frac{\text{(Line 13) x (Col. F) x (Col. K) x (Abbrev. Fac.)}}{\text{(Col. E., Step 2A Worksheet Abbrev.)}}$$

NOTE: The Abbreviation Factor is the same as used in Step 2A Worksheet. (See Step 2A Worksheet Instructions for further explanation.)

Col. M. Plot "X" coordinate (Column J), and "Y" coordinate (Column L) on appropriate F-Chart and enter monthly percentage of total load carried by solar. (F-Charts in DATA portion of this section.)

Col. N. Calculate solar load (monthly and yearly) in Btu:

(Col. E, Step 2A Worksheet Abbrev.) x (Col. M)

Line 14. Calculate yearly percentage of total load carried by solar:

$$\frac{\text{(Col. N. Abbreviation TOTAL)}}{\text{(Col. E, Step 2A Worksheet Abbrev. TOTAL)}}$$

COMPARING COLLECTOR CURVES

More than one collector may be plotted on a performance graph as illustrated by Figure 5-15. This provides for a direct performance comparison. For convenience, the intercepts and slopes of the two curves are listed on the graph. These may be referred to and used in the thermal calculations which will be discussed in this section.

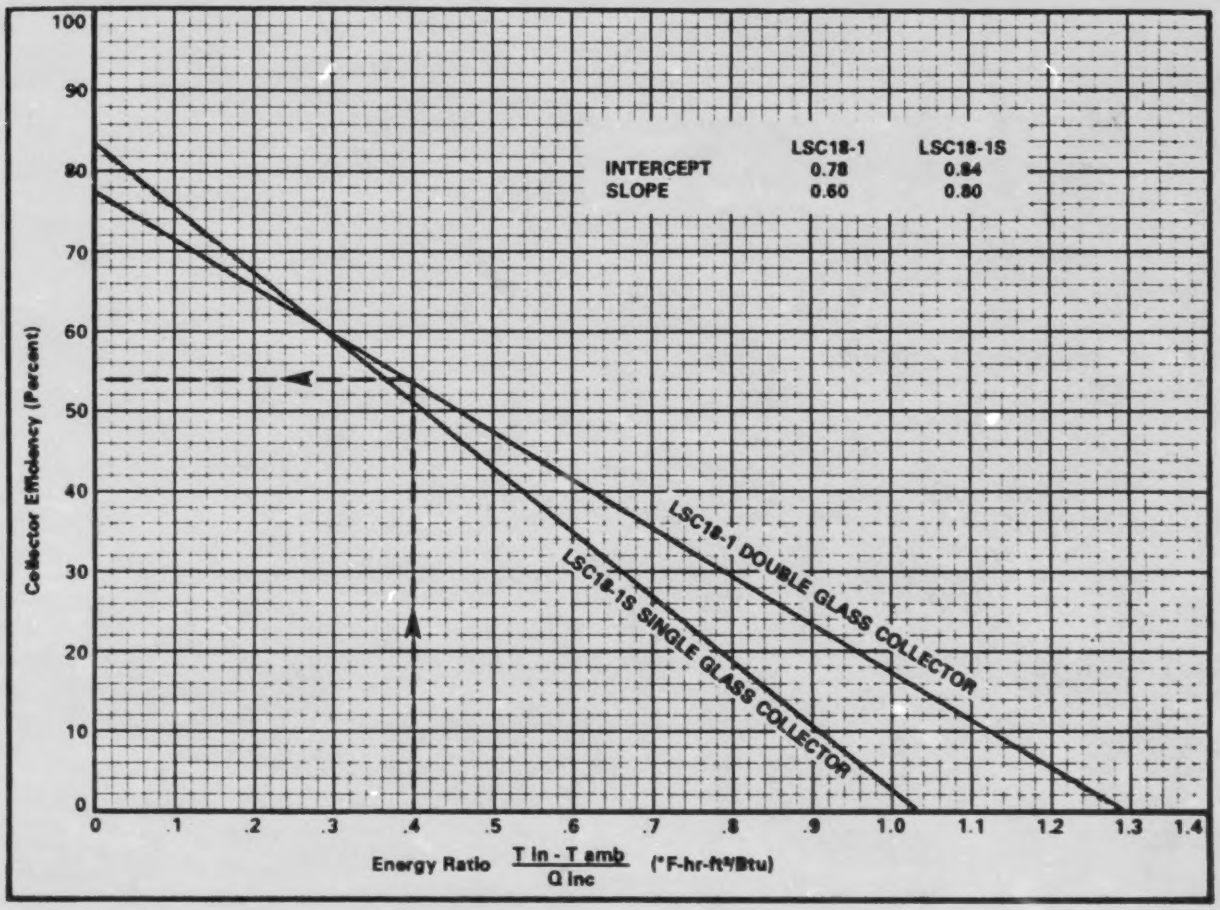


Figure 5-15: Comparing Collector Performance Charts.

DETERMINING BTU CAPACITY OF COLLECTOR

The performance curve can be used to determine Btu capacity of a collector when the amount of available incident solar radiation (Q inc) is know. The Energy Ratio Scale formula (See Fig. 5-15) is used. The available radiation in this example is 250 Btu/hr-ft^2.

Example Calculation

Assume there is a fluid inlet temperature (T_{in}) of 110° F and an ambient temperature (T_{amb}) of 10° F. Subtracting ambient temperature from inlet temperature results in a 100° temperature difference. The temperature difference (100) divided by the solar radiation (250) equals .40. Refer to the collector performance chart. Follow the example line from .40 of the bottom scale vertically to where it intersects the curve of the LSC18-1 Double Glass Collector. Read horizontally to the Collector Efficiency Scale from that point of intersection. This line indicates an efficiency of 54%. Thus, the collector is 54% efficient under the conditions used in the example. The Btu capacity of the collector in this instance would be 250 Btu/hr/ft^2 multiplied by 54%, or 135 Btu/hr-ft^2.

HEAT ENERGY REMOVAL

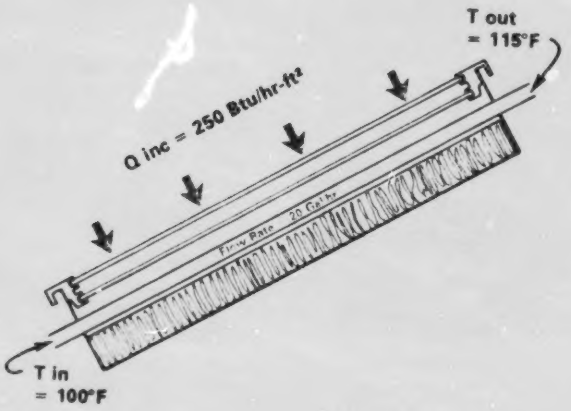
The rate heat energy is removed from the collector affects collector performance and efficiency. The radiant energy falling on the collector absorber plate, which is converted to heat energy, must be continually removed by the working fluid. Otherwise, the plate temperature rises until heat losses equal energy collected. The more rapidly the energy can be removed from the absorber plate, the more efficient the collector will operate.

Two factors affect the rate of heat removal. One is the volume of the fluid circulating across the plate. The other is the temperature of the fluid when it enters the collector. It is a combination of the weight of the fluid and the difference in entering and leaving temperature (temperature rise) of the fluid that determines the amount of heat removed.

For example, 20 gallons of water at an inlet temperature of 100° F is circulated through the collector in one hour. See Figure 5-16. The water temperature leaving the collector is 115° F. The weight of water is 8.3 pounds. Thus, 20 gallons times 8.3 pounds, or 166 pounds, of water is circulated through the collector in one hour. Since the water temperature is increased 15°, the rate of heat removal from the collector plate is 15° times 166 pounds, or 2,490 Btu per hour.

HEAT REMOVAL EFFICIENCY

The efficiency of heat collection and removal is determined by dividing the rate of heat removal by the total radiation intercepted. In the previous example, the rate of heat removal was 2,490 Btuh. The total radiation intercepted is the available incident radiation (use 250 Btu/hr-ft^2) times the effective absorber area (use 15.4 square feet), or 3,850 Btu per hour. The efficiency of heat collection and removal is 2,490 Btu/hr. divided by 3,850 Btu/hr., or 64.7%. See Figure 5-16.



HEAT REMOVAL RATE

A. FLUID FLOW RATE = 20 GAL/HR
B. WEIGHT OF GALLON OF FLUID = 8.3 LBS
C. TOTAL FLUID FLOW = 20 GAL x 8.3 LBS = 166 LBS/HR
D. TEMPERATURE RISE OF FLUID = T OUT - T IN =
115°F - 100°F = 15°F
E. HEAT REMOVED BY FLUID = 15°F x 166 LBS/HR =
2490 BTU/HR

HEAT REMOVAL EFFICIENCY

A. AVAILABLE INCIDENT RADIATION = Q INC =
250 BTU/HR - FT²
B. EFFECTIVE ABSORBER AREA = 15.4 FT²
C. TOTAL RADIATION INTERCEPTED = 15.4 x 250 =
3850 BTU/HR
D. EFFICIENCY OF HEAT REMOVAL = $\frac{2490\ \text{BTU}}{3850\ \text{BTU}}$ = 64.7%

Figure 5-16

INSTRUCTIONS FOR STEP 2C WORKSHEET

NOTE: Solar percentage of the total load for two additional estimated collector areas can be figured on this worksheet.

Line 15. Choose estimated total collector areas that differ from Line 7 of Step 2B Worksheet.

NOTE: Complete balance of worksheet as directed. Upon completion, select the desired collector area and proceed with sizing the remainder of the components.

STEP 2C. SOLAR LOAD PERCENTAGES FOR OTHER COLLECTOR AREAS WORKSHEET

(Instructions for worksheet on next page.)

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 215.6 sq. ft.
(14 COLLECTORS)

Line 16. Collector Area Factor:
(Line 15 215.6)
(Line 7, Step 2B Worksheet 277) = ... .78

Col. O MONTH	Col. P "X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	Col. Q "Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	Col. R SOLAR PERCENTAGE (from F-Chart)	Col. S SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	.90	.31	.24	4.70 x 10^6
Feb.	1.01	.41	.32	5.10 " "
Mar.	1.19	.51	.39	5.41 " "
April	1.93	.90	.62	4.67 " "
May	3.39	1.71	.95	3.93 " "
June	6.08	3.42	1.00	2.10 " "
July	7.24	4.31	1.00	1.76 " "
Aug.	6.58	3.80	1.00	1.95 " "
Sept	4.24	2.33	1.00	3.01 " "
Oct.	2.29	1.10	.72	4.59 " "
Nov.	1.28	.48	.39	4.72 " "
Dec.	.98	.30	.22	3.84 " "
			YEARLY TOTAL	45.78 " "

Line 17. Yearly Solar Load Percentage:
(Col. S Abbreviation TOTAL 45.78 x 10^6)
(Col. E, Step 2A Worksheet Abbreviation TOTAL 105.84 x 10^6) = ... 43.25 %

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 338.8 sq. ft.
(22 COLLECTORS)

Line 16. Collector Area Factor:
(Line 15 338.8)
(Line 7, Step 2B Worksheet 277) = ... 1.22

Col. O MONTH	Col. P "X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	Col. Q "Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	Col. R SOLAR PERCENTAGE (from F-Chart)	Col. S SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	1.42	.49	.36	7.05 x 10^6
Feb.	1.57	.63	.47	7.50 " "
Mar.	1.87	.81	.57	7.90 " "
April	3.01	1.39	.84	6.33 " "
May	5.29	2.67	1.00	4.14 " "
June	9.52	5.36	1.00	2.10 " "
July	11.32	6.73	1.00	1.76 " "
Aug.	10.28	5.94	1.00	1.95 " "
Sept.	6.64	3.65	1.00	3.01 " "
Oct.	3.59	1.72	.94	6.00 " "
Nov.	2.01	.76	.56	6.78 " "
Dec.	1.54	.46	.34	5.94 " "
			YEARLY TOTAL	60.46 " "

Line 17. Yearly Solar Load Percentage:
(Col. S Abbreviation TOTAL 60.46 x 10^6)
(Col. E, Step 2A Worksheet Abbreviation TOTAL 105.84 x 10^6) = ... 57.1 %

COLLECTOR SIZING - SA/L METHOD

INTRODUCTION

SA/L is another, more versatile method for approximating the size of the collector array than the LC factor method used in Step 1A Worksheet. Where the LC factor method limits the collector array to be sized according to three fixed solar load percentages (25%, 50% and 100%), the SA/L solar load percentages range from 0% to 100%. In addition, SA/L can also be used to determine solar load percentage when the design heat load and size of the collector array are shown.

Two factors are important in the concept of SA/L. They are the January heating load for a structure and the radiation falling on a horizontal surface at a specific location in January. The concept of SA/L correlates those two factors with the percent of heat load that would be carried by solar for the entire year.

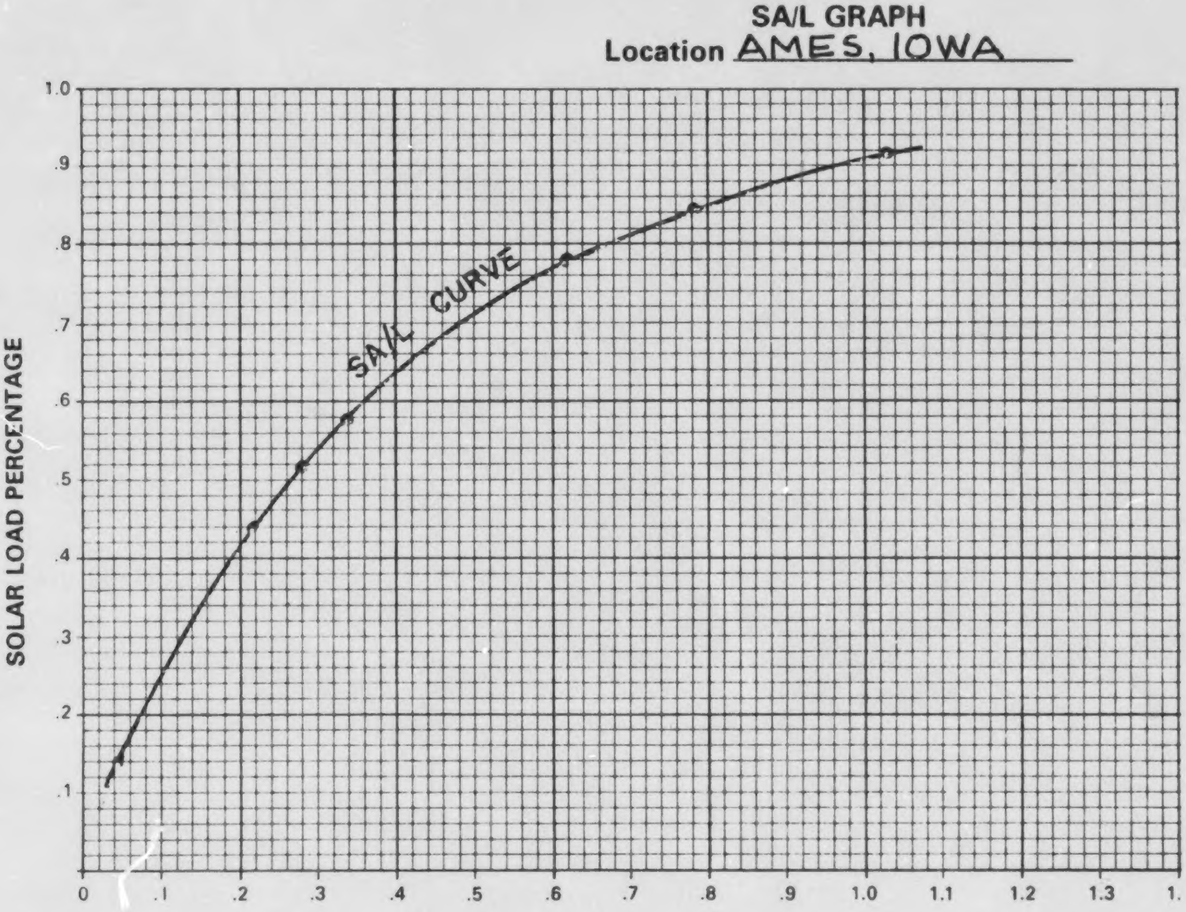
In the SA/L formula the "S" stands for the solar radiation on a horizontal surface at a specific location for the entire month of January. The "A" stands for the collector area. The "L" stands for the total January heat load of a typical residence, including domestic water heating.

The SA/L Method of collector sizing also incorporates five variables that should match those already used in Steps 2A and 2B Worksheets. These variables are losted below (specific variables for Ames example system are in parenthesis):

1. Cover Type collector used (Lennox double glass);
2. Tilt angle of collector array is equal to latitude of location + 10° (50°);
3. Daily domestic hot water usage (80/gal.day);
4. Mean temperature of cold supply water to the water heater (55° F);
5. Setpoint of conventional water heater (140° F).

The initial step in the SA/L method is developing an SA/L curve for a typical residential solar application in a specific location or area (See SA/L Curve Development Worksheet on the next page). In developing the curve for the first time, the designer should complete Steps 1A, 2A, 2B, 2C and the SA/L Curve Development Worksheets. Although more steps are involved in initially using SA/L than the LC factor method, once the curve has been established for a certain area it can be reused over and over again. If the designer wishes to use SA/L after the curve has been developed, he should follow this procedure:

1. Skip Step 1A Worksheet;
2. Complete SA/L Short Form Worksheet to qualify the customer, and
3. Complete Collector Sizing - Long Form (Steps 2A, 2B, and possibly 2C) Worksheets.



SA/L Factor

V-S-148

COLLECTOR SIZING — SA/L CURVE DEVELOPMENT WORKSHEET

Location AMES, IOWA

Line 1. Find "S", the solar radiation falling on a horizontal surface at a specific location in January:
(first line in first column of AMES WEATHER AND RADIATION TABLE 641) x 31 (days in January) = ... 19,871 Btu/mo.-ft²

Line 2. Find "A", the Collector Area:
(from Line 7, Step 2B Worksheet) .... 277 sq. ft.

Line 3. Find "L", Total Heat Load for January, including domestic water heating from January entry, Col. E, Step 2A Worksheet .... 19.59×10^6 Btu/mo.

Line 4. Find SA/L Factor:
S (Line 1 19,871) x A (Line 2 277)
L (Line 3 19.59×10^6) = ... .28 SA/L Factor

Line 5. Percentage of Heating Load to be carried by solar (from Line 14, Step 2B Worksheet) .... 50.8 %

Line 6. Refer to the SA/L graph on next page. Plot the point made by the intersection of SA/L Factor (Line 4 .28) and percentage of heating load to be carried by solar (Line 5 50.8).

Line 7. Find SA/L Factors for other collector areas figured in Step 2C Worksheets. Also enter the corresponding Solar Load Percentage figured for each collector area. NOTE: Four additional collector areas were figured in order to plot a more accurate SA/L curve (see additional Step 2C Worksheets following the SA/L graph).

S (Line 1 19,871) x A (Line 15, Step 2C Worksheet 215.6)
L (Line 3 19.59×10^6) = ... .22 SA/LFactor
Solar Heat Load Percentage (from line 17, Step 2C Worksheet) .... 43.25 %

S (Line 1 19,871) x A (Line 15, Step 2C Worksheet 338.8)
L (Line 3 19.59×10^6) = ... .34 SA/L Factor
Solar Heat Load Percentage (from Line 17, Step 2C Worksheet) .... 57.1 %

S (Line 1 19.871) x A (Line 15, Step 2C Worksheet 46.2)
L (Line 3 19.59×10^6) = ... .05 SA/L Factor
Solar Heat Load Percentage (from Line 17, Step 2C Worksheet) .... 13.68 %

S (Line 1 19.871) x A (Line 15, Step 2C Worksheet 616)
L (Line 3 19.59×10^6) = ... .62 SA/LFactor
Solar Heat Load Percentage (from Line 17, Step 2C Worksheet) .... 76.57 %

S (Line 1 19.871) x A (Line 15, Step 2C Worksheet 770)
L (Line 3 19.59×10^6) = ... .78 SA/L Factor
Solar Heat Load Percentage (from Line 17, Step 2C Worksheet) .... 84.17 %

S (Line 1 19.871) x A (Line 15, Step 2C Worksheet 1001)
L (Line 3 19.59×10^6) = ... 1.02 SA/L Factor
Solar Heat Load Percentage (from Line 17, Step 2C Worksheet) .... 90.92 %

Line 8. Plot the corresponding SA/L Factor and Solar Heat Load percentage points calculated from Line 7 on SA/L graph on next page.

Line 9. Complete the SA/L curve by connecting the points already plotted on the SA/L graph.

STEP 2C. SOLAR LOAD PERCENTAGES FOR OTHER COLLECTOR AREAS WORKSHEET

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 46.2 sq. ft.
(3 COLLECTORS)

Line 16. Collector Area Factor:
(Line 15 46.2)
(Line 7, Step 2B Worksheet 277) = ... .17

Col. O	Col. P	Col. Q	Col. R	Col. S
MONTH	"X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	"Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	SOLAR PERCENTAGE (from F-Chart)	SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	.20	.07	.06	1.18 x 10^6
Feb.	.22	.09	.08	1.28 " "
Mar.	.26	.11	.10	1.39 " "
April	.42	.19	.16	1.21 " "
May	.74	.37	.31	1.28 " "
June	1.33	.75	.56	1.18 " "
July	1.58	.94	.67	1.18 " "
Aug.	1.43	.83	.61	1.19 " "
Sept.	.92	.51	.41	1.23 " "
Oct.	.50	.24	.20	1.28 " "
Nov.	.28	.11	.10	1.21 " "
Dec.	.21	.06	.05	0.87 " "
			YEARLY TOTAL	14.48 " "

Line 17. Yearly Solar Load Percentage:
(Col. S Abbreviation TOTAL 14.48x10^6)
(Col. E, Step 2A Worksheet Abbreviation TOTAL 105.84x10^6) = ... 13.68 %

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 616 sq. ft.
(40 COLLECTORS)

Line 16. Collector Area Factor:
(Line 15 616)
(Line 7, Step 2B Worksheet 277) = ... 2.22

Col. O	Col. P	Col. Q	Col. R	Col. S
MONTH	"X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	"Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	SOLAR PERCENTAGE (from F-Chart)	SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	2.57	.89	.58	11.36 x 10^6
Feb.	2.86	1.15	.72	11.48 " "
Mar.	3.40	1.47	.85	11.78 " "
April	5.48	2.53	1.00	7.54 " "
May	9.63	4.86	1.00	4.14 " "
June	17.32	9.75	1.00	2.10 " "
July	20.60	12.25	1.00	1.76 " "
Aug.	18.71	10.81	1.00	1.95 " "
Sept.	12.07	6.64	1.00	3.01 " "
Oct.	6.53	3.13	1.00	6.38 " "
Nov.	3.66	1.38	.85	10.29 " "
Dec.	2.80	.84	.53	9.25 " "
			YEARLY TOTAL	81.04 " "

Line 17. Yearly Solar Load Percentage:
(Col. S Abbreviation TOTAL 81.04x10^6)
(Col. E, Step 2A Worksheet Abbreviation TOTAL 105.84x10^6) = ... 76.57 %

STEP 2C. SOLAR LOAD PERCENTAGES FOR OTHER COLLECTOR AREAS WORKSHEET

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 770 sq. ft.
(50 COLLECTORS)

Line 16. Collector Area Factor:
$$\frac{\text{(Line 15 } 770\text{)}}{\text{(Line 7, Step 2B Worksheet } 277\text{)}} = \ldots\ldots\ldots\ldots\ 2.78$$

Col. O MONTH	Col. P "X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	Col. Q "Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	Col. R SOLAR PERCENTAGE (from F-Chart)	Col. S SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	3.22	1.11	.68	13.32×10^6
Feb.	3.59	1.45	.83	13.24 " "
Mar.	4.25	1.83	.94	13.03 " "
April	6.87	3.17	1.00	7.54 " "
May	12.07	6.09	1.00	4.14 " "
June	21.68	12.20	1.00	2.10 " "
July	25.80	15.55	1.00	1.76 " "
Aug.	23.44	13.54	1.00	1.95 " "
Sept.	15.12	8.31	1.00	3.01 " "
Oct.	8.17	3.92	1.00	6.38 " "
Nov.	4.59	1.72	.96	11.62 " "
Dec.	3.50	1.06	.63	11.00 " "
			YEARLY TOTAL	89.09 " "

Line 17. Yearly Solar Load Percentage:
$$\frac{\text{(Col. S Abbreviation TOTAL } 89.09 \times 10^6\text{)}}{\text{(Col. E, Step 2A Worksheet Abbreviation TOTAL } 105.84 \times 10^6\text{)}} = \ldots\ldots\ldots\ 84.17\ \%$$

Line 15. Estimated Effective Absorber Area of Collector Array (sq. ft.) .... 1001 sq. ft.
(65 COLLECTORS)

Line 16. Collector Area Factor:
$$\frac{\text{(Line 15 } 1001\text{)}}{\text{(Line 7, Step 2B Worksheet } 277\text{)}} = \ldots\ldots\ldots\ldots\ 3.61$$

Col. O MONTH	Col. P "X" COORDINATE: (Line 16) x (Col. J, Step 2B Worksheet)	Col. Q "Y" COORDINATE: (Line 16) x (Col. L, Step 2B Worksheet)	Col. R SOLAR PERCENTAGE (from F-Chart)	Col. S SOLAR LOAD (Btu/mo.): (Col. E, Step 2A Worksheet Abbreviation) x (Col. R)
Jan.	4.19	1.44	.79	15.48×10^6
Feb.	4.66	1.88	.94	14.99 " "
Mar.	5.52	2.38	1.00	13.86 " "
April	8.92	4.12	1.00	7.54 " "
May	15.67	7.91	1.00	4.14 " "
June	28.16	15.85	1.00	2.10 " "
July	33.50	19.93	1.00	1.76 " "
Aug.	30.43	17.58	1.00	1.95 " "
Sept.	19.64	10.79	1.00	3.01 " "
Oct.	10.61	5.09	1.00	6.38 " "
Nov.	5.96	2.24	1.00	12.10 " "
Dec.	4.55	1.37	.74	12.92 " "
			YEARLY TOTAL	96.23 " "

Line 17. Yearly Solar Load Percentage:
$$\frac{\text{(Col. S Abbreviation TOTAL } 96.23 \times 10^6\text{)}}{\text{(Col. E, Step 2A Worksheet Abbreviation TOTAL } 105.84 \times 10^6\text{)}} = \ldots\ldots\ldots\ 90.92\ \%$$

COLLECTOR SIZING — SA/L SHORT FORM WORKSHEET

Part I. Use When Collector Area Is Known And Solar Heat Load Percentage For That Area Is Unknown.

Line 1. Desired Collector Area, "A" .... 277 sq. ft.
NOTE: This area is arbitrarily determined by the designer and/or homeowner. (18 COLLECTORS)

Line 2. Find SA/L Factor:
S (Line 1 of SA/L Curve Worksheet 19,871) x A (Line 1 277)
L (Line 3 of SA/L Curve Worksheet 19.59 x 10⁶) = .. .28 SA/L Factor

Line 3. Find Solar Heat Load Percentage:
(Refer to SA/L graph. Find SA/L Factor (Line 2) on SA/L Factor scale of graph. Move straight upward from SA/L Factor until intersecting SA/L curve. Then move straight left until intersecting Solar Heat Load Percentage scale.)
Solar Percentage = ... 51 %

Part II. Use When Desired Solar Heat Load Percentage Is Known, And Collector Area Needed To Obtain That Percentage Is Unknown.

Line 4. Desired Solar Load Percentage .... 50 %
NOTE: This percentage is arbitrarily determined by the designer and/or homeowner.

Line 5. Find SA/L Factor:
(Refer to SA/L graph. Find Solar Percentage (Line 4) on Solar Percentage scale of graph. Move straight right until intersecting SA/L Factor curve. Then move from intersection of curve straight down until intersecting SA/L Factor scale.)
SA/L Factor = ... .27 SA/L Factor

Line 6. Find Collector Area, "A", from the following formula:
SA/L Factor (Line 5 .27) x L (Line 3 of SA/L Curve Worksheet 19.59 x 10⁶)
S (Line 1 of SA/L Curve Worksheet 19,871) = 266 sq. ft.

Line 7. Find Number of Collectors Needed:
(Line 6 266)
Collector Effective Absorber Area 15.4 (from Collector EHB data) = 17.3 collectors
OR 18

As a general rule, the most cost-effective collector array will be one that provides 60 - 70% of the annual load (solar fraction - .6 - .7). The sizing method used in the example can provide a means for decision-making that is much faster than the cumbersome manual f-Chart method. If a computer is available, the f-Chart program is a very quick and acceptable accurate method.

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF THE STORAGE SYSTEM

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF THE STORAGE SYSTEM

With a heat supply that is both variable and interruptable, it is logical to collect solar energy when it is abundant, use it as needed, and store any extra heat for later use, either at night or on sunless days. In the early days of solar heating, the practice was to install a very large storage capacity that was planned to supply heat for several days without sun. (See Figure 6-1). Today, the trend is toward a more modest storage capacity.

CONCEPTS OF HEAT STORING MATERIALS

All substances - whether they are in a solid, liquid, or gaseous phase (form) - are capable of absorbing heat. Any given substance has a specified relationship

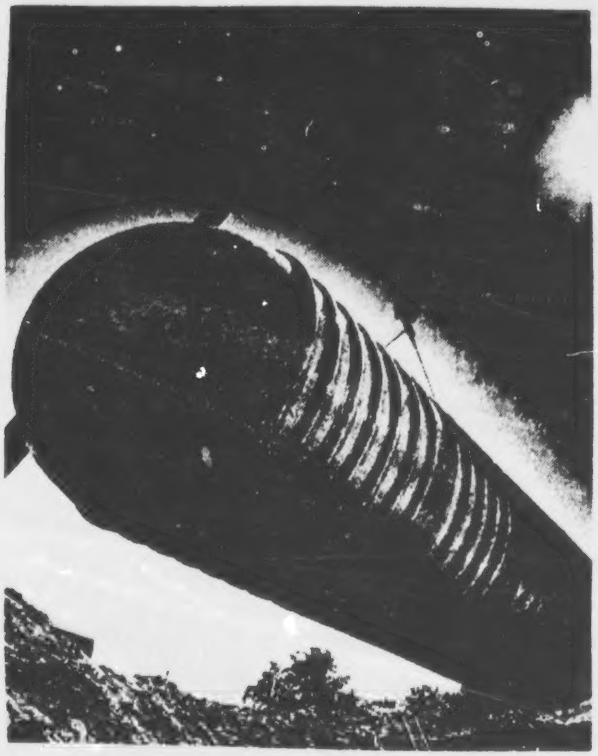


Figure 6-1: Fiberglass tank for "low" temperature water storage. Residential needs would be satisfied by a much smaller tank.

to water in its ability to absorb heat. Water has a specific heat of 1.0 Btu per pound for each degree of Fahrenheit rise in temperature. This means that a cubic foot of water weighing 62 pounds (or 8.34 lb/gallon) will have absorbed 62 Btu's if its temperature rises 1° F. For example, a 2000 gallon water storage tank with water heated from 90° to 100° F, would absorb 2000 gallons x 8.34 lb/gal. x 1 Btu/lb x (100° F - 90° F), or 166,800 Btu's.

Rock, such as granite, is another common material that can be used for heat storage. The specific heat of rock is about 0.20 Btu/lb per °F. A cubic foot of lightly packed two-inch rocks weighs about 100 pounds. Therefore, a cubic foot of rocks would store 20 Btu's when raised 1° F.

These heat storage capacity specifications, for water or rock, help to explain the volumetric space requirements for fluid storage mediums. In a one cubic foot container, five times as many Btu's can be stored if water, rather than rock, is the storage medium. The solar heating system that functions by warming water or rock is classified as a sensible heat storage system.

The alternative to the sensible heat storage system is the phase change storage system. An example of this type of heat storing practice may be seen with paraffin wax. During the day, the wax melts as it absorbs solar heat. During the night, it cools and freezes as it releases heat to the air. Melting wax has about four times the heat absorbing qualities as water and, theoretically, could store a comparable amount of heat in 1/4 the storage space required for water.

When the storage medium undergoes a change of phase; that is, changes from a solid to a liquid and back again, the term "heat of fusion" storage is sometimes applied. Almost any substance can be changed from solid-to-liquid-to-gas by the addition of heat as shown in Figure 6-2. In this diagram, the solid substance is ice. As heat is applied and absorbed by the ice, temperature is raised to 32° F, whereby the ice melts. It absorbs 144 Btu's for each pound of melting ice. Further heating increases the temperature of the liquid from 32° to 212° F. At 212° F, a second phase change occurs as the water turns to steam - absorbing 970 Btu's per pound in the process. If the process were reversed, the same number of Btu's would have to be removed as the steam changes to water and then to ice.

With certain waxes and salts, however, the solid-to-liquid heat of fusion can be made to occur at more elevated temperatures, between 90° and 120° F. Therefore, a larger amount of heat associated with phase change is gained or lost at a temperature suitable for direct use in space heating. (Figure 6-3).

Phase change materials have some disadvantages. Substances such as waxes and

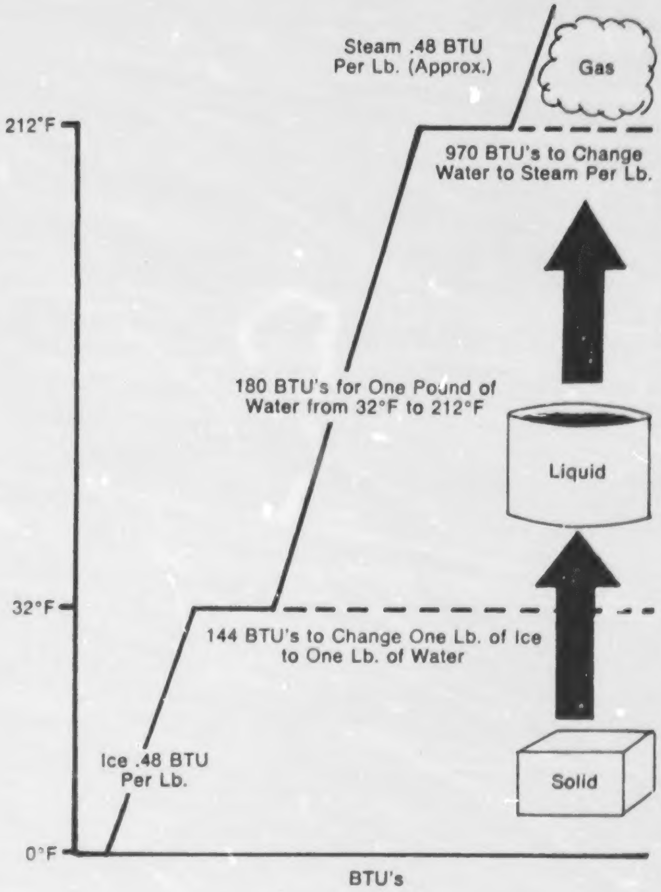


Figure 6-2: Phase Change from Solid to Liquid to Gas.

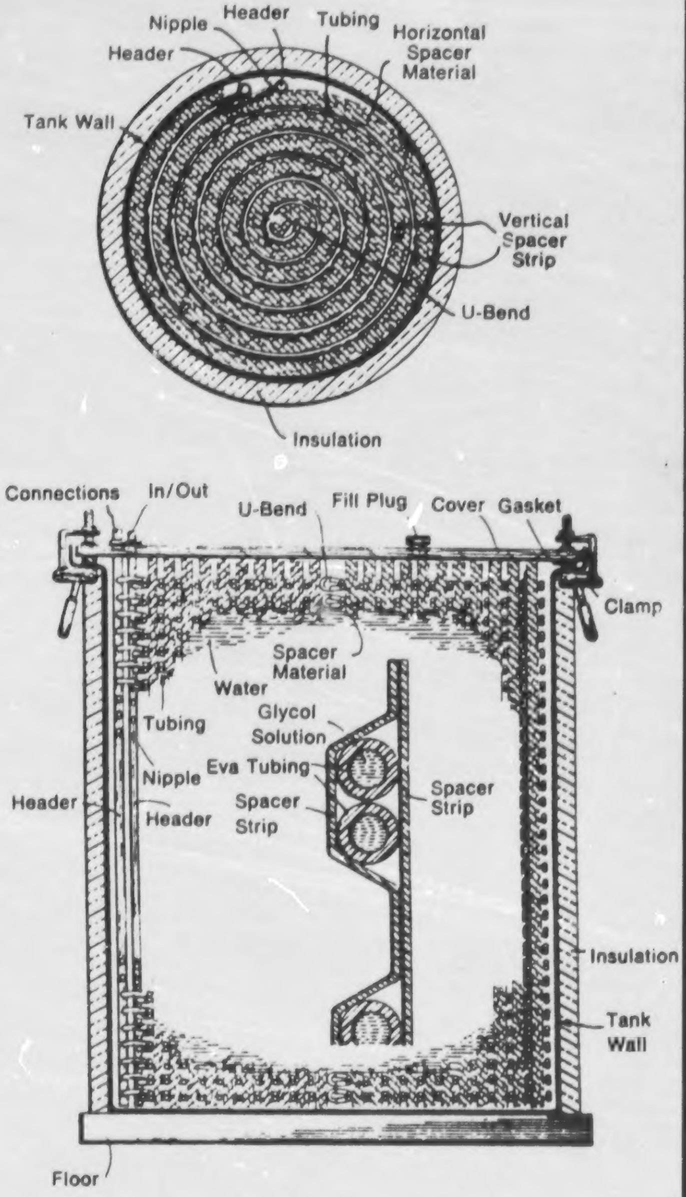

Nipple
Header
Tubing
Horizontal
Spacer
Material
Header
Tank Wall
Vertical
Spacer
Strip
U-Bend
Insulation
Connections
In/Out
U-Bend
Fill Plug
Cover
Gasket
Clamp
Spacer
Material
Water
Glycol
Solution
Tubing
Eva Tubing
Nipple
Spacer
Strip
Header
Header
Spacer
Strip
Insulation
Tank
Wall
Floor

Figure 6-3: Phase change storage Unit. Plastic Mat Heat Exchanger sits in container filled with an inorganic salt hydrate.

salts are quite expensive at the present time. Also, many must be replaced regularly because of chemical decomposition. In addition, the design of phase change containers is complicated by the need for a large heat transfer surface. Hence, much of the space saving feature of using a phase change material is lost, because the container must be enlarged for practical considerations.

The question of what storage medium to utilize is based on:

1. the ability of a substance to accept heat,
2. the cost of the amount required for a given heat demand, and
3. the availability of the material.

With this information in mind, the remainder of the lessons in this course will be limited to sensible heat storage practices using water and rocks.

CONCEPTS OF HEAT STORAGE SIZING

In the first section of this Module, the nature of different heat storage materials was discussed. The purpose of that information was to provide an awareness of some of the possible mediums to use for storing heat. In this section, the essential considerations of sizing air and liquid storage units will be explained.

Solutions to the problem of adequate heat storage relate to both the building's heat loss or "energy demand" and the amount of energy that the collectors can provide. In other words, the volume of space for storage depends on the size of the building and the size of the collectors.

The interdependence of storage size upon collector size is illustrated in Figure 6-4. Time of day is represented on the horizontal scale and energy in Btu's per hour is provided in the vertical scale. The top horizontal line represents the heat loss of a building, as might be calculated using NESCA's Manual J. The wavy horizontal line indicates the actual hourly heat loss for the hypothetical building on a particular day, when outdoor conditions were much milder than the design day. From just before 9 a.m. and just after 3 p.m., there are two curves for the amount of solar energy collected by two different sizes of collector arrays. The shaded area indicates the amount of energy used to heat the house between 9 a.m. and 3 p.m. The net amount of collected energy available for storage is the clear area above the shaded region. The smaller collector obviously has less excess energy available for storage than the larger collector. This factor is always true, regardless of the size of the storage unit that is installed. In other words, a storage unit can be too big.

Figure 6-4 illustrates only one operating mode. There are other possibilities. On a very mild day, all the energy collected could go to storage, and conversely, on a near-design day, there may be no excess collectable energy for storage.

A storage capacity that is too large could also affect the temperature of the storage medium, such as the water in a tank. There is a minimum supply water temperature for effective heating with a fan coil, panel, or baseboard unit. An excessively large storage tank may take days to be heated to a useful temperature if the collector to storage capacities are extreme (small collector, large tank).

Conversely, a storage unit that is too small may become overheated and the operating temperature in the collector may be excessive. This would decrease the collector's efficiency.

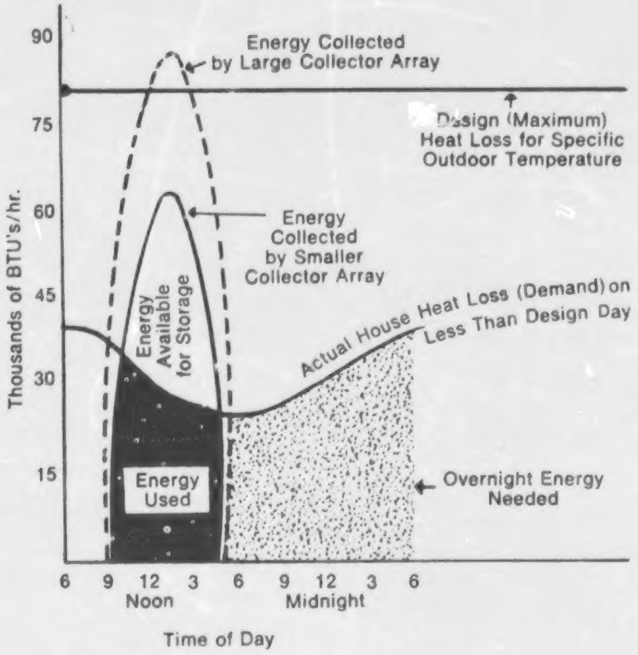


Figure 6-4: Solar Energy Availability and Needs.

Figure 6-5 represents a trend curve noting the effect of storage size on the contribution of solar heat to the total heat supplied to a building. With no storage, a solar system would, perhaps, contribute no more than 40% of the heating needs (assuming a fixed size of collector). The impact of providing a very large storage capacity would not significantly increase the percent contribution much above 80%, which could be achieved by a moderately sized storage unit. Cost consideration is still another factor that can limit the size of storage provided. Doubling the cost of the storage facility to gain two or three percent more solar contribution is not generally considered to be worthwhile.

What is a reasonable, moderate storage capacity for residential type solar heating installations? Designers do not all agree! But generally, storage of from one to two gallons of water per square foot of installed collector has been successful. For rock storage, from one-half to one cubic foot of rocks per square foot of collector is quite effective. Proprietary solar assisted heat pump applications may require slightly different storage ratios, and the manufacturer must be consulted regarding these.

The other side of the collector storage problem is - as was mentioned - the thermal size of the building.

To size conventional heating equipment, a design heat loss calculation is made. This is based on local design outdoor conditions and recommended indoor

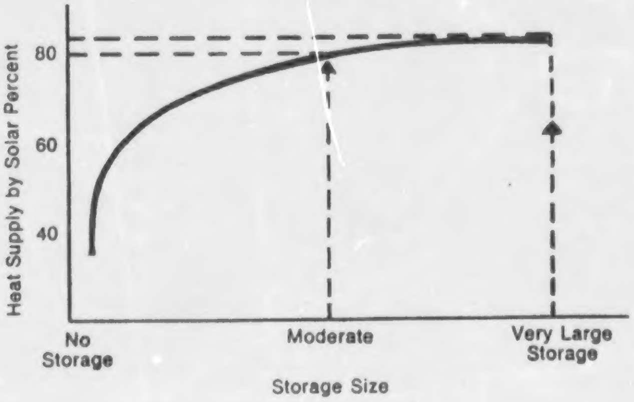
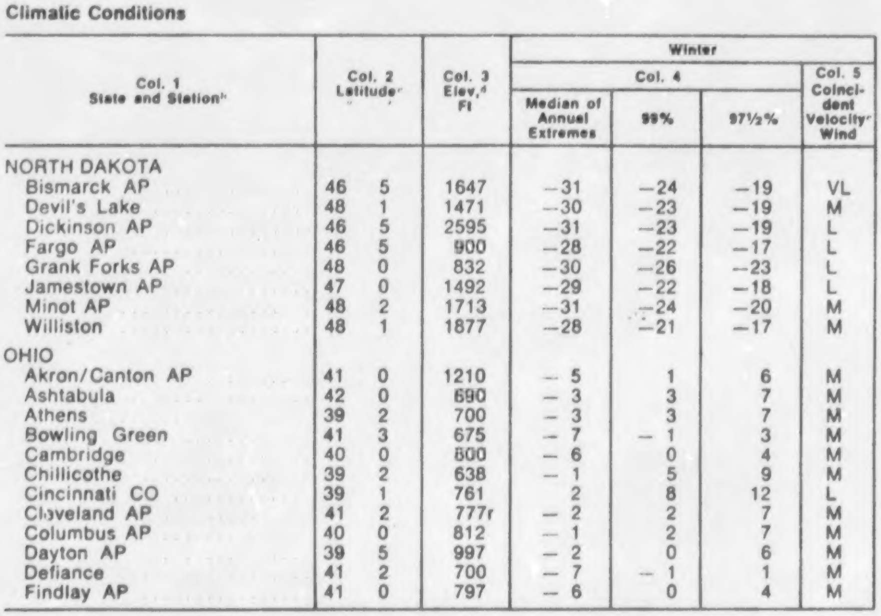


Figure 6-5: Effect of Storage Size on Heat Contributed by Solar Energy.

design temperature. The outdoor design temperature in a given locality is not the lowest ever recorded temperature, but rather a somewhat higher temperature based on frequency of occurence. For example, Figure 6-6 illustrates a section of the ASHRAE Table of climatic data for the United States and Canada. Note that three outdoor temperatures are listed for each city - median of extremes, 99% and 97-1/2%. Median value is the middle value of the coldest temperature recorded each year for 30 years. The 99% and 97-1/2% of the total winter operating hours for December, January and February. In other words, out of 2160 hours in those three months, the temperature might fall below the listed value for each city for 21.6 hours for 99% and 543 hours for 97-1/2 percent columns.

Since a design heat loss calculation is an estimate of the maximum or near maximum load of a structure in terms of Btu's per hour, it cannot be used to estimate the overnight, daily or monthly Btu requirements of the building. For example, a house with a design heat loss of 50,000 Btu will not require 50,000 x 24 hours, or 1,200,000 Btu's per day, unless the outdoor temperature for that particular day was at the design value for 24 hours. Since outdoor temperature is usually a variable through the day, a mean (average) daily temperature must be used to estimate daily Btu requirements.



Climatic Conditions

Col. 1 State and Station[b]	Col. 2 Latitude[c]		Col. 3 Elev.[d] Ft	Winter			
				Col. 4			Col. 5 Coincident Velocity[e] Wind
	°	′		Median of Annual Extremes	99%	97½%	
NORTH DAKOTA							
Bismarck AP	46	5	1647	−31	−24	−19	VL
Devil's Lake	48	1	1471	−30	−23	−19	M
Dickinson AP	46	5	2595	−31	−23	−19	L
Fargo AP	46	5	900	−28	−22	−17	L
Grank Forks AP	48	0	832	−30	−26	−23	L
Jamestown AP	47	0	1492	−29	−22	−18	L
Minot AP	48	2	1713	−31	−24	−20	M
Williston	48	1	1877	−28	−21	−17	M
OHIO							
Akron/Canton AP	41	0	1210	− 5	1	6	M
Ashtabula	42	0	690	− 3	3	7	M
Athens	39	2	700	− 3	3	7	M
Bowling Green	41	3	675	− 7	− 1	3	M
Cambridge	40	0	800	− 6	0	4	M
Chillicothe	39	2	638	− 1	5	9	M
Cincinnati CO	39	1	761	2	8	12	L
Cleveland AP	41	2	777r	− 2	2	7	M
Columbus AP	40	0	812	− 1	2	7	M
Dayton AP	39	5	997	− 2	0	6	M
Defiance	41	2	700	− 7	− 1	1	M
Findlay AP	41	0	797	− 6	0	4	M

Figure 6-6: Climatic Conditions for North Dakota and Ohio

Mean daily temperature is 1/2 the sum of the maximum and minimum temperature for a specific day, see Figure 6-7. For example, assume that in a 24 hour period, the outdoor temperature ranged from a high of 40° F to a low of 18° F. The mean daily temperature would be 40 + 18 divided by 2, or 29° F. The estimated daily Btu requirements are as follows:

$$\frac{\text{design heat loss}}{\text{design temperature difference}} \times \text{Mean daily temperature difference} \times 24 \text{ hours} = \text{daily Btu requirements}$$

If the indoor temperature is 70° F, and the outdoor design temperature is 0° F, then, using a 50,000 Btu house, the formula is:

$$\frac{50{,}000}{(70 - 0)} \times (70 - 29) \times 24 = 702{,}857 \text{ Btu's per day.}$$

As is generally acknowledged, almost any building has internal heat released from lights, cooking, and other activities; there is also some solar heat gain directly through windows. In an effort to account for this heat contribution,

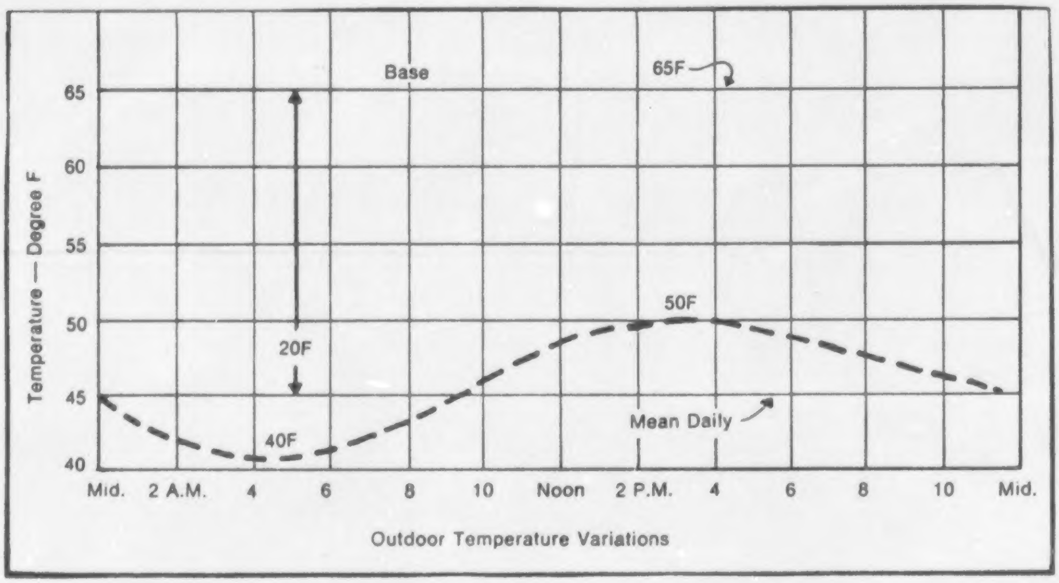


Figure 6-7: Idealized Outdoor Temperature Fluctuations Throughout the Day. Mean Daily Temperature is Half the Sum of the Maximum and Minimum Temperature for a Specific 24-Hour Period.

and to estimate more precisely what actual Btu's the heating system must supply, the mean daily temperature difference is usually adjusted by changing the indoor temperature from 70° F to 65° F. Thus, the formula would be:

$$\frac{50{,}000}{(70 - 0)} \quad x \quad (65 - 29) \quad x \quad 24 \quad = \quad 617.142 \text{ Btu's per day}$$

The value (65-29) is commonly called Degree Days. Monthly and yearly Degree Days have been published for many years for given cities. A sample tabulation is shown in Figure 6-8.

If, for example, the hypothetical building was located in Dayton, Ohio, which has 5,622 annual degree days, the estimated annual energy requirements would be:

$$\frac{50{,}000}{(70 - 0)} \quad x \quad 5{,}622 \text{ x } 24 \quad = 96{,}377{,}140 \text{ Btu's per season.}$$

Average Monthly and Yearly Degree Days

State	Station		Avg. Winter Temp.[d]	July	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	Mar.	Apr.	May	June	Yearly Total
	New York (Kennedy)	A	41.4	0	0	36	248	564	933	1029	935	815	480	167	12	5219
	Rochester	A	35.4	9	31	126	415	747	1125	1234	1123	1014	597	279	48	6748
	Schenectady	C	35.4	0	22	123	422	756	1159	1283	1131	970	543	211	30	6650
	Syracuse	A	35.2	6	28	132	415	744	1153	1271	1140	1004	570	248	45	6756
N.C.	Asheville	C	46.7	0	0	48	245	555	775	784	683	592	273	87	0	4042
	Cape Hatteras		53.3	0	0	0	78	273	521	580	518	440	177	25	0	2612
	Charlotte	A	50.4	0	0	6	124	438	691	691	582	481	156	22	0	3191
	Greenboro	A	47.5	0	0	33	192	513	778	784	672	552	234	47	0	3805
	Raleigh	A	49.4	0	0	21	164	450	716	725	616	487	180	34	0	3393
	Wilmington	A	54.6	0	0	0	74	291	521	546	462	357	96	0	0	2347
	Winston-Salem	A	48.4	0	0	21	171	483	747	753	652	524	207	37	0	3595
N.D.	Bismarck	A	26.6	34	28	222	577	1083	1463	1708	1442	1203	645	329	117	8851
	Devils Lake	C	22.4	40	53	273	642	1191	1634	1872	1579	1345	753	381	138	9901
	Fargo	A	24.8	28	37	219	574	1107	1569	1789	1520	1262	690	332	99	9226
	Williston	A	25.2	31	43	261	601	1122	1513	1758	1473	1262	681	357	141	9243
Ohio	Akron-Canton	A	38.1	0	9	96	381	726	1070	1138	1016	871	489	202	39	6037
	Cincinnati	C	45.1	0	0	39	208	558	862	915	790	642	294	96	6	4410
	Cleveland	A	37.2	9	25	105	384	738	1088	1159	1047	918	552	260	66	6351
	Columbus	A	39.7	0	6	84	347	714	1039	1088	949	809	426	171	27	5660
	Columbus	C	41.5	0	0	57	285	651	977	1032	902	760	396	136	15	5211
	Dayton	A	39.8	0	6	78	310	696	1045	1097	955	809	429	167	30	5622
	Mansfield	A	36.9	9	22	114	397	768	1110	1169	1042	924	543	245	60	6403
	Sandusky	C	39.1	0	6	66	313	684	1032	1107	991	868	495	198	36	5796
	Toledo	A	36.4	0	16	117	406	792	1138	1200	1056	924	543	242	60	6494
	Youngstown	A	36.8	6	19	120	412	771	1104	1169	1047	921	540	248	60	6417

Figure 6-8: Degree Days

If an estimate is made of the January energy requirement for the building, use the January degree days for Dayton, which are 1,097. The estimated January load would be:

$$\frac{50{,}000}{(70 - 0)} \times 1097 \times 24 = 18{,}805{,}714 \text{ Btu's for January.}$$

As a first approximation, it is also possible to divide the January total by 31 days to arrive at a "typical" day's energy need in January. Thus:

$$\frac{18{,}805{,}714}{31} = 606{,}636 \text{ Btu's per January day.}$$

If the solar collectors operate for 8 hours, then 16/24 of the all day load would be approximately the Btu requirement needed from storage - provided a full 16 hour storage requirement was expected. Thus:

$$\frac{16}{24} \times 606{,}636 = 404{,}424 \text{ Btu's overnight load.}$$

How much storage would be required to satisfy the estimated overnight load? If water was used and a 30° F rise in storage temperature occurs, then:

$$404{,}424 = 8.34 \text{ lb./gal.} \times 1 \times 30^{\circ}\text{ F} \times \text{gallons}$$

$$\text{or gallons} = \frac{404{,}424}{8.34 \times 1 \times 30}$$

If rock storage is required and a 70° F rise in storage temperature occurs, then:

$$404{,}424 = 20 \text{ Btu/cu ft F} \times 70^{\circ}\text{ F} \times \text{cubic foot}$$

$$\text{or cu ft} = \frac{404{,}424}{20 \times 70} = 288 \text{ cu. ft. of rocks}$$

In the above calculations, 30° F rise for water storage and 70° F rise for rock storage were arbitrarily chosen. Particularly in the case of design storage temperature for water, where a liquid-to-air heat exchanger is to be used, the storage temperature must be carefully selected.

As a general rule, air temperature leaving the room supply register should be approximately 20° (or more) warmer than the desired room temperature. If this minimum Δ t of the return air is to be achieved, an efficient heat exchanger must be selected, so that the air temperature leaving the exchanger will approach (or exceed) the desired minimum Δ t of the air. Heat exchanger design and selection procedures will be discussed in the module following this one.

Example: Room design temperature is 70° F, requiring a minimum 90° F air temperature at the supply registers. Assuming a 2° F temperature loss in the duct tystem, the air must leave the heat exchanger at 92° F (minimum). Thus, the temperature of the liquid in the heat exchanger must be higher than 92° F. The efficiency of the heat exchanger enters into the calculations here. Assuming an efficiency of 70% (air temperature rises 70% of the difference between return air temperature and exchanger liquid temperature), the temperature of the liquid would have to be at least 101.4° F.

$$\text{Liquid Temperature} = \frac{(92 - 70)}{.7} + 70 = \frac{22}{.7} + 70 = 101.4^\circ \text{ F.}$$

In the above example, 101° F would be the minimum storage temperature and the maximum required temperature to meet the load could be calculated:

$$Q = Mc\,\Delta t; \qquad t = \frac{Q}{Mc}$$

where the mass (M) is fixed by storage container size. Conversely, if Δ t is fixed by collector performance, then mass would be the variable, and

$$M = \frac{Q}{c\,\Delta t}$$ would apply.

It may not be economical to provide complete overnight storage. At this point, however, one should have an understanding of the calculations required to determine building needs and storage capacity.

PLACEMENT OF HEAT STORAGE UNITS

Solar heat storage units may be placed either inside or outside the building they are serving. They can be located either below grade or above grade. The rationale for a storage unit location is that it be placed in a low value area of the structure or building site.

Units placed below grade inside the building could be in a basement or crawl space. This area would be close to most of the other components in the heating system. It would be protected from moisture and cold. Any excessive radiation energy coming in and/or heat loss radiating from the storage container could be directed into the heated space as needed, or it could be vented to the outside through a duct system.

A low value space should also be sought when placing the heat storage unit above grade inside the structure, such as a small room or attic. Because of the weight of the container and the storage media (pebbles or water), extensive foundation reinforcement would be necessary. In one case, to serve the needs of a thermosyphon system used primarily to service household requirements for domestic hot water, the heat storage tank must be about two feet above the collector. That would mean using an inconvenient, but low value, attic installation.

Units placed outside and above ground would be exposed to whatever minimum temperature that would be reached in a given geographical location. They would also create a problem aesthetically.

Units placed outside and below ground pose some problems also. They would need to be buried below the frostline, so that they would not be affected by pressure from the frozen soil. Water or air leaks would be difficult to locate. Digging may be required before servicing, unless the container was in some sort of a pit. In this case, an access cover would provide a way into the storage unit.

There are problems of storage unit placement associated with each type of solar heating system. Not only is this true for the architects and contractors when constructing new facilities, but in addition, inconvenient when existing buildings are retrofitted with solar heat. Some previously used living areas must be converted to non-usable space to accommodate heating system components.

ADVANTAGES AND DISADVANTAGES FOR STORING HEATED LIQUIDS

There are advantages and disadvantages to each and every type of solar heating system, regardless of the type fluid (liquid or air) for which the system was designed. Advantages are that:

1. Water is a relatively abundant and inexpensive medium,
2. water absorbs and emits heat readily, and
3. space requirements for water storage unit placement is the least for any of the common heat storage mediums used presently.

However, there are some disadvantages. First, the system must be insulated against freezing. (It may be too costly to use antifreeze for all the water in storage). Second, boiling may occur in the collector, and a ventilating valve may have to be installed to remove steam pressure if overheating occurs. Third, circulating the water throughout the system would require a pump and the system would be vulnerable to leakage. This would create the need for periodic inspections, troubleshooting, and maintenance. A fourth problem is that components made of aluminum, copper or brass, and iron, which are threaded or joined directly to each other will cause an electrolytic action. Such action will form initially at the connection, spread through the components, and reduce the flow rate by constricting the pipe diameter. (See Figure 6-9 on the following page).

Over a long period of time, corrosion can cause total erosion of pipe or fitting, and a leak will occur. As a result, pressure in the system would drop, antifreeze and inhibitors would be lost, and the system would stop functioning.

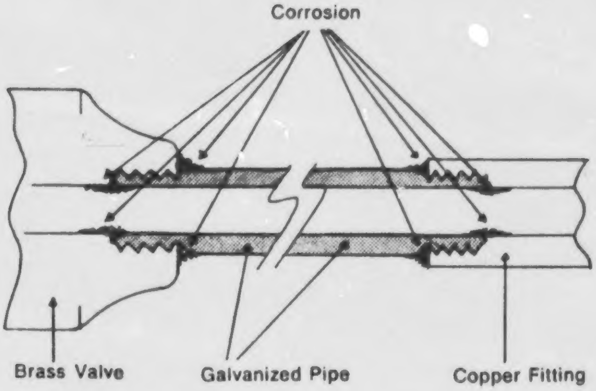


Figure 6-9: Electrolytic Corrosion

The rate of corrosion can be affected in two ways:

1. by the amount and type of minerals in the liquid, and
2. by the temperature of the fluid, since heat increases the rate of corrosion.

One form of controlling corrosion is to add a corrosion inhibitor into the system. Another procedure is to use lengths of plastic pipe or dielectric unions with plastic bushings between the incompatible metal components.

A fifth problem for a liquid storage system is accessability. Buried or enclosed storage tanks and/or pipes would be difficult to inspect for leakage. If a pipe or tank froze and ruptured, or if chemical corrosion caused a failure in the system, repairs could be extensive. This aspect applies not only to the replacement of various components of the solar heating system, but also to the possible need to redecorate walls, ceiling or floors where damage may have occurred.

WATER TANK MATERIALS

There are several different materials commonly used for liquid storage systems. Concrete, steel, fiberglass and butyl-lined concrete block are four of the materials frequently used. Selection should be made based on the required capacity and location of the storage unit. Figure 6-10 represents a typical tank installation.

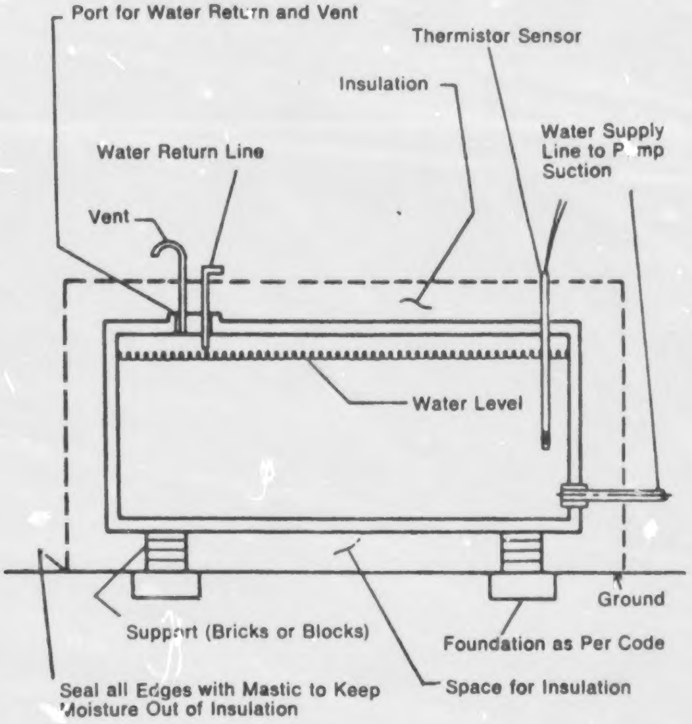

Port for Water Return and Vent
Thermistor Sensor
Insulation
Water Supply Line to Pump Suction
Water Return Line
Vent
Water Level
Ground
Support (Bricks or Blocks)
Foundation as Per Code
Seal all Edges with Mastic to Keep Moisture Out of Insulation
Space for Insulation

Figure 6-10: Typical Above-Ground Liquid Storage Tank Installation

Concrete Tanks

Concrete tanks are probably the most durable. These units cannot corrode or be punctured. A waterproof sealer on the inside surface or a waterproof liner will free the container from possible seepage problems. Another engineering problem to contend with is the weight and size of the tank to be installed. A crane or other heavy equipment is needed when the tank is set in place. This is particularly difficult when placing a tank in the basement of an existing structure. Watertight pipe connections are also difficult to achieve, without using a high-quality caulking compound, such as silicone.

Fiberglass Tanks

A fiberglass tank especially designed for use with high temperature liquids is probably the most advisable system for water storage. The tank must be of a quality for protection against rupturing at a high temperature, such as the 212° F boiling point of water. Tanks used for gasoline storage may not meet this requirement. Care must be taken to determine safe pressure and temperature limits for the tanks. The tanks must be well insulated after they are installed, and must be checked for leaks if the solar heating system is to operate satisfactorily.

Glass-Lined or Galvanized Steel Tanks

Glass-lined or galvanized tanks have been found extremely serviceable for many years. Galvanized tanks, however, may become corroded and inserviceable before those which are glass-lined. The glass-lined tanks may have their expected years of usage shortened by careless installation. Excessive pressure in securing connections can crack or break the inflexible glass and allow more rapid deterioration. It may be advantageous to use several smaller tanks because they require less floor space in a basement and in a retrofit, smaller tanks may be installed more readily.

ADVANTAGES AND DISADVANTAGES OF PEBBLE-BED HEAT STORAGE UNITS

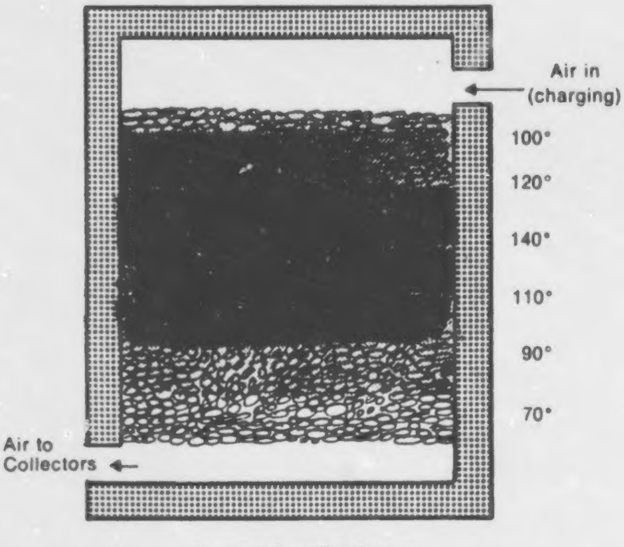
Heat storage units for air-circulating solar heating systems are called pebble-beds. They contain a large volume of pebbles (granite or other clean crushed rock about 3/4 to 1-1/2 inches in diameter). During the heat circulating cycle, air is forced through the inlet; then it filters through the pebbles and into ducts, where it is transferred to the space to be heated. When the rocks need to be recharged with heat, the air movement is reversed and hot air is drawn in through the outlet, and heats the pebbles. A four foot deep bed of pebbles is considered the maximum for a minimum air pressure drop between the inlet and outlet.

Pebble-bed containers have been found very efficient. They are considered by many to be better than the water and/or phase change systems. They are simple to construct, by:

1. Laying cement blocks and filling the holes with concrete;
2. Pouring concrete into a form, or
3. Fabricating them from wall stud material, plywood, and insulation.

The pebbles should be hand-shoveled or chuted into the unit, to insure even distribution and reduce packing or side wall stress. For example, a cylinder three feet by eighteen feet holds about 12,000 pounds of carefully sized, crushed rock.

They have a high level of thermal stratification. Heat inside the container exists in layers. When the pebble bed is being charged with heated air from the collectors, most of the heat is transferred to the rocks within a few feet of the bed, and the air leaving the bottom of the storage bin is essentially at 70° F. A representative temperature profile in a storage bin is shown in Figure 6-11. This profile obviously changes with the length of the charging time.



Thermal Stratification

Figure 6-11: Temperature "Layers" in Pebble Bed Storage

When flow is reversed to extract heat from storage, the hot end of the bin heats the leaving air to within a few degrees of the actual rock temperature for effective utilization of stored energy.

Cubic pebble-beds (Figure 6-12) are preferred because heat has a natural tendency to rise. This style of unit is normally about five feet high. Due to the weight of the container and its contents, these units must have rather extensive foundations so that the force of the weight will not cause the unit to crack. If a space three feet high is the maximum available (in a crawl space, for example), then a box using a horizontal air flow, as shown in Figure 6-13, is adequate.

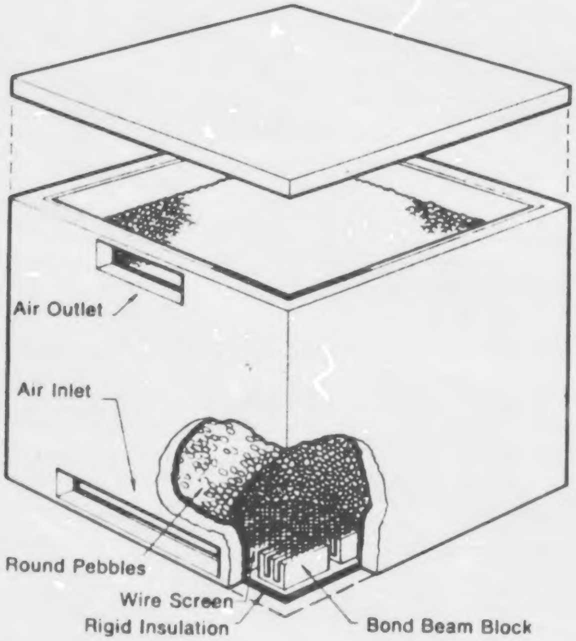

Air Outlet
Air Inlet
Round Pebbles
Wire Screen
Rigid Insulation
Bond Beam Block

Figure 6-12: Cube Type Pebble Bed Storage.

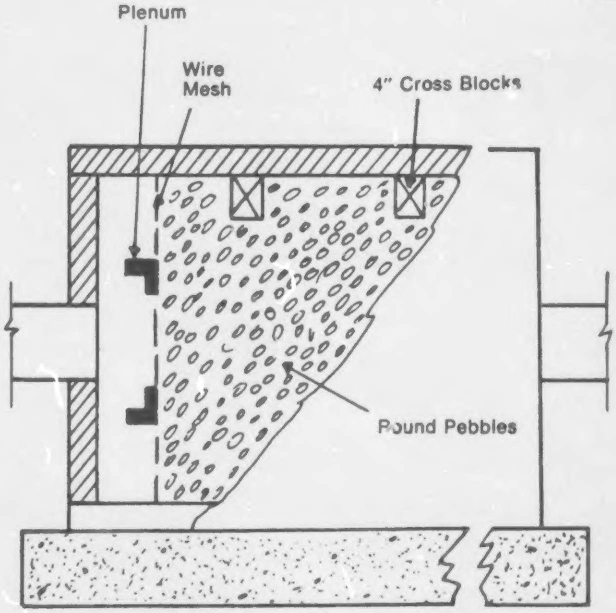

Plenum
Wire Mesh
4" Cross Blocks
Round Pebbles

Figure 6-13: Horizontal Pebble-Bed Storage.

Tank Material

Tank materials most likely to be used in water storage are steel, aluminum, concrete, and plastics. A number of different modular and built-in-place tanks are shown in Figures 6-14 through 6-19. Steel tanks should be lined with a material such as butyl rubber to prevent internal corrosion from the water. Any material used as a lining should have a long life, since replacement may be quite difficult. The lining materials must also withstand high temperatures that occur in the storage tank. Concrete tanks may not require lining, depending on quality, but require additional reinforcing and sealing of joints because of temperature stresses. Most plastic tanks currently available will not withstand the temperatures needed in solar heating and cooling systems. Therefore, special composition tanks will be required.

Tank Shape

A spherical tank provides the least surface area per unit volume of storage. It is cheapest to insulate. A shape that deviates from this, such as a long slender cylinder, required more tank insulation material.

A spherical tank is also structurally advantageous. However, fabrication and support of spherical storage tanks are more difficult than for cylindrical tanks. Thus, the most practical shape is a cylindrical tank having a diameter-to-width ratio of nearly one. Precast concrete tanks could be cylindrical or rectangular. Built-in-place reinforced concrete tanks would likely be rectangular, because forming is easier.

Tank Insulation

Insulation of a storage tank is important to conserve the collected heat. The bottom as well as the sides of the tank should be insulated. Tank bottom insulation must be accomplished prior to installation of the tank. There are several possibilities for doing this. One is to rest the tank on rigid insulation foam pads. Another is to rest the tank on closely spaced two-by-six inch boards and insulate between them.

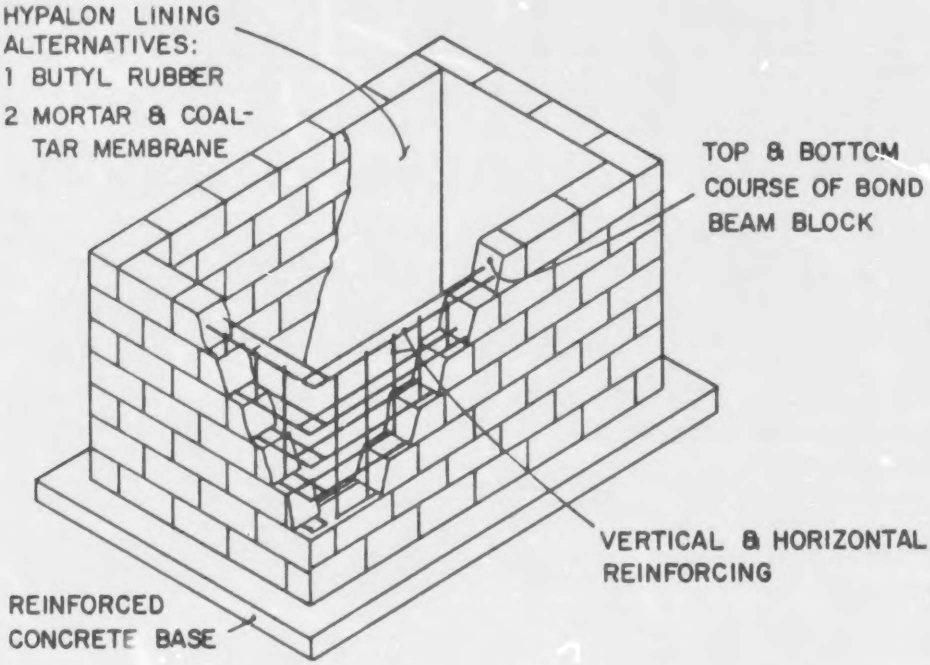

HYPALON LINING
ALTERNATIVES:
1 BUTYL RUBBER
2 MORTAR & COAL-TAR MEMBRANE
TOP & BOTTOM COURSE OF BOND BEAM BLOCK
VERTICAL & HORIZONTAL REINFORCING
REINFORCED CONCRETE BASE

Figure 6-14: Reinforced Concrete Block Tank

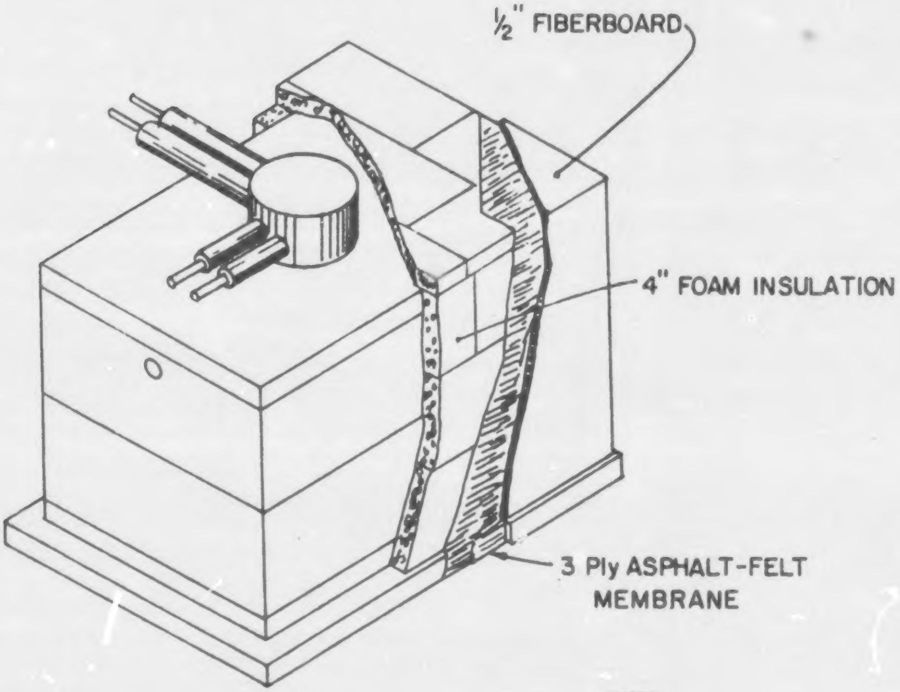

½" FIBERBOARD
4" FOAM INSULATION
3 Ply ASPHALT-FELT MEMBRANE

Figure 6-15: Precast Sectional Utility Vault

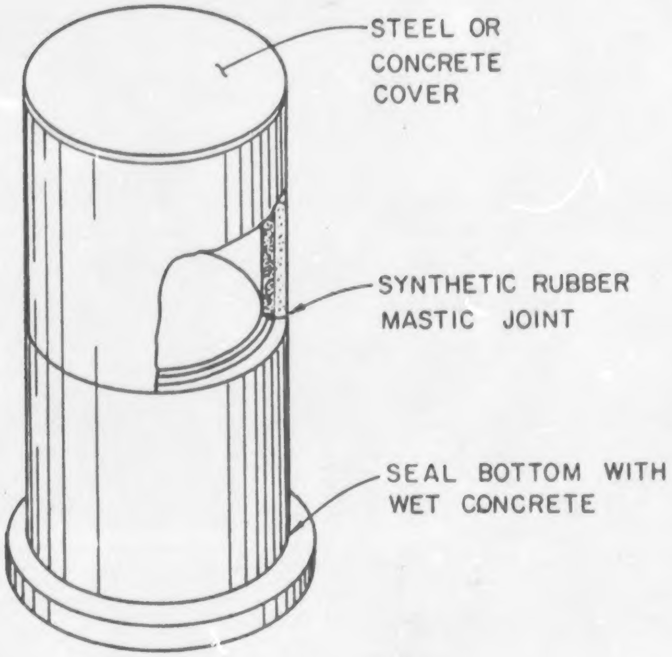
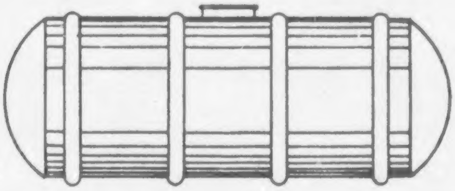
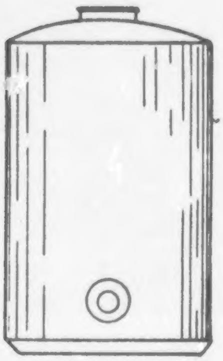

STEEL OR CONCRETE COVER
SYNTHETIC RUBBER MASTIC JOINT
SEAL BOTTOM WITH WET CONCRETE

Figure 6-16: Precast Concrete Storm Drain Pipe Tank



UNDERGROUND STORAGE TANK



ABOVE GROUND STORAGE TANK

Figure 6-17: Fiberglass Tanks

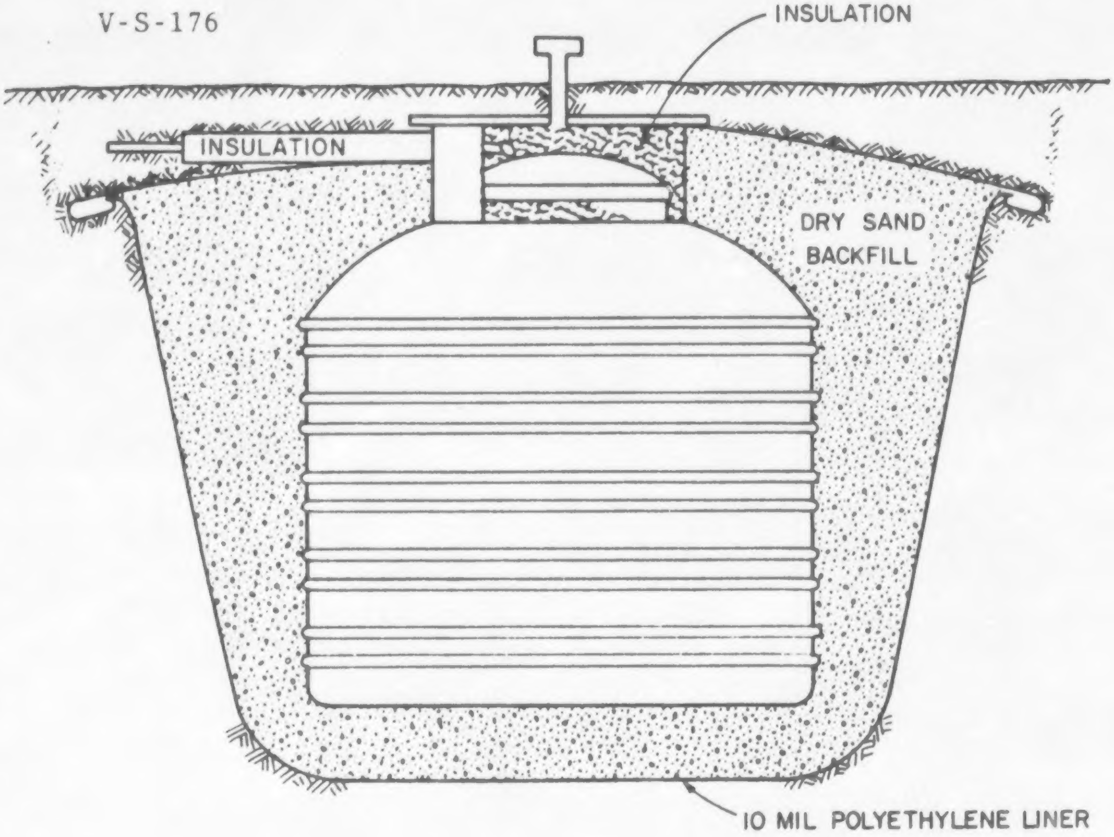

INSULATION
INSULATION
DRY SAND BACKFILL
10 MIL POLYETHYLENE LINER

Figure 6-18: Fiberglass Septic Tanks - Various Shapes - 500-1000 Gal.

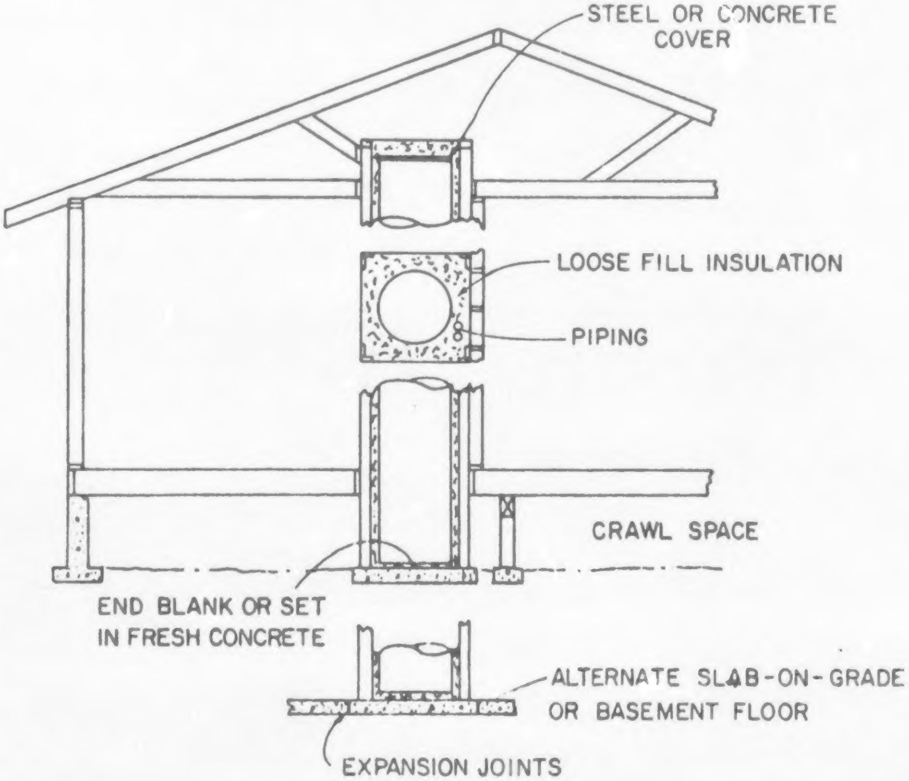

STEEL OR CONCRETE COVER
LOOSE FILL INSULATION
PIPING
CRAWL SPACE
END BLANK OR SET IN FRESH CONCRETE
ALTERNATE SLAB-ON-GRADE OR BASEMENT FLOOR
EXPANSION JOINTS

Figure 6-19: Vertical Tanks from Pipe Sections

Several different approaches can be used to insulate. One is to wrap the tank in conventional insulating material, and another is to use a spray foam type of insulation. Insulation of at least R-23 is recommended for inside and R-30 for outside tank placements. A type of insulation that will not absorb moisture should be selected in outside locations or other areas where water or moisture may be a problem.

It is also desirable to enclose the tank along with the associated components, such as the hot water heater and heat exchangers, in a vented, insulated room. This isolates the tank and other heat-producing subsystems from the rest of the house.

Tank Location

If the tank is located within the building, there is some loss of living area. Usually the most desirable location for the tank is in the basement. In this case, the tank is easily accessible for repairs. In other cases, such as retrofit applications, or where there is no basement, other locations must be found. Alternatives are the garage or outside the house, either above ground or buried.

Tank Stratification

There is little stratification in a hot water storage tank. The difference in temperature of the water between the top and bottom of the tank is about 5° F during normal operation in 1000-gallon tanks. Stratification can be enhanced to some degree by the introduction of baffles to prevent convection and mixing. However, it is questionable that the gain is worth the expense. Another possibility is to use multiple tanks, but this adds to the expense of the system.

Two major advantages can be gained by stratification. One is higher temperature water delivered to the heating coils, and the other is colder water delivered to the collectors. Higher temperatures to the fan coils could result in smaller sizes and colder temperatures to the collector, resulting in increased collection efficiencies.

Piping to the Tanks

The inlet pipe to the storage tank from the collector should be located toward the top of the tank, and the outlet to the collector should be at the bottom of the tank. The outlet from the tank for house heating and cooling should be toward the top of the tank, where the tank is the hottest, and the return should be toward the bottom of the tank. Vented tanks should be provided with a make-up water line leading to the bottom of the tank with a float control valve.

Storage Tank Size

Studies have shown that for most locations in the United States, the storage tank should be sized to hold from 1.5 to 2.5 gallons of water per square foot of collector area. A small storage tank will have higher average temperatures and hence greater heat losses. However, high storage temperatures are desirable for air-conditioning applications, since the cut-off temperature for an absorption air-conditioning unit is about 170° F. A large storage tank will have lower average temperatures and may not be able to provide direct heating of the house. For most residential applications and locations, the system performance is relatively insensitive within the 1.5 to 2.5 gallon per square foot range.

ROCK BED STORAGE

Container Arrangement

A rock bed storage bin can be constructed with wood. Stud walls with one-half inch plywood on both sides and 3.5 inches of R-11 insulation placed between the studs form an adequate storage container. A plenum must be provided on the top and bottom to distribute air flow evenly over the container cross-section, as shown in Figure 6-20.

The bottom plenum is constructed by supporting expanded wire mesh on concrete blocks placed about 1.5 inches apart. The rocks are then placed on top of the wire mesh and the bin is filled to within a foot of the top of the rocks. The space at the top of the bin above the rocks forms the top plenum.

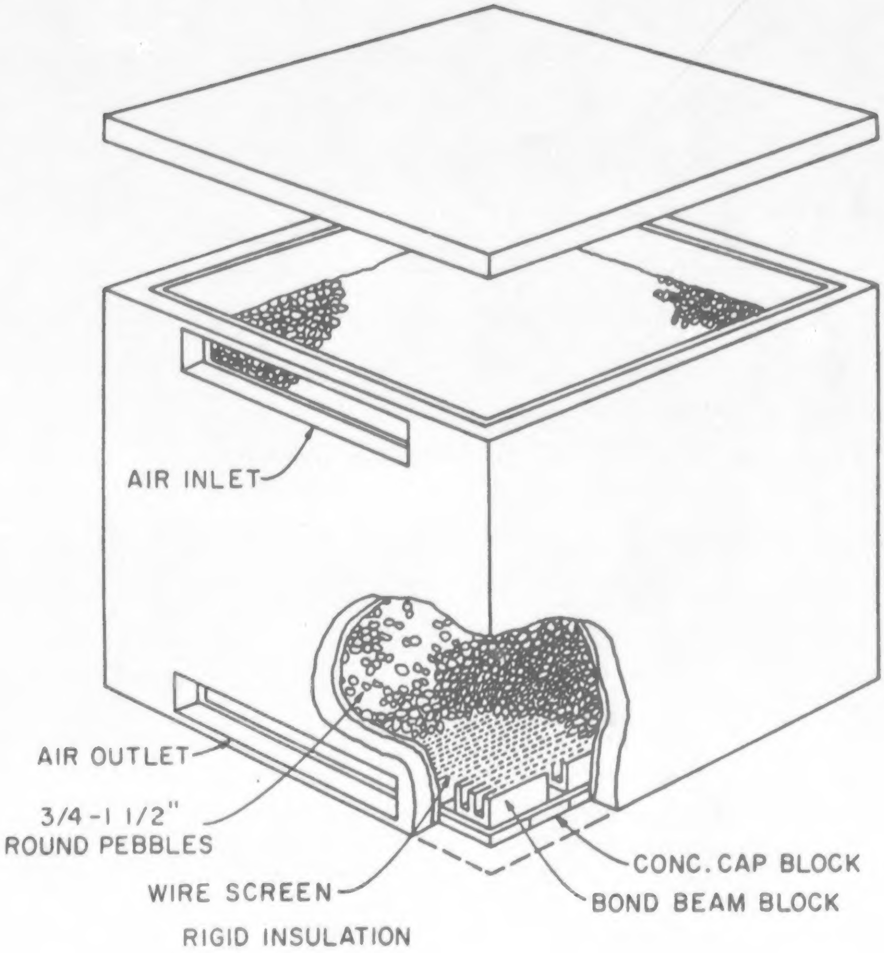

AIR INLET
AIR OUTLET
3/4-1 1/2"
ROUND PEBBLES
WIRE SCREEN
RIGID INSULATION
CONC. CAP BLOCK
BOND BEAM BLOCK

Figure 6-20: Rock Bed Heat Storage Unit

The bin should be sealed before the rocks are placed to prevent air leakage. This is accomplished by caulking the joints with an epoxy or other suitable heat-resistant compound. Butyl rubber gasket or other heat-resistant material can be used to form the seal for the top lid. Air temperatures are nominally 150° F, but higher temperatures of 180° to 190° F are sometimes reached. Thus, sealant materials should be selected to within the higher temperatures.

Container Size and Shape

The rock bed storage should be sized to provide 50 to 100 pounds of rock per square foot of collector. For normal rock densities and for .75-to-1.5-inch sizes, this is equivalent to one-half to one cubic foot per square foot of collector.

Ideally a rock bed should have a short distance from inlet to outlet, with a large cross-sectional area perpendicular to the direction of air flow. With a large cross-sectional area, the air velocity is low and, coupled with a short travel path through the rock bed, the pressure drop is small. The smaller pressure drop results in lower fan power.

The rock bed must be deep enough to permit stratification. A minimum depth of 2.5 feet is recommended. In order to have adequate storage, however, the volume of the rock bed must be large. To avoid a large cross-sectional area with consequent displaced floor area, a larger depth may be used. When rock beds are constructed in the building, a depth of about five feet is allowable.

Figure 6-21 shows representative temperature profiles as they develop throughout the day for a typical 4.5-foot-high rock bed storage. In this figure, the bed is assumed to be fully discharged at the beginning of the day. From this figure it can be seen that a 2.5-foot-high bed will cause high outlet temperatures at the bottom after 1:00 p. m.

When storage is not fully discharged by morning, the temperature profiles of Figure 6-21 would be displaced to the right by the end of the day. A five-foot rock bed depth can therefore be advantageous.

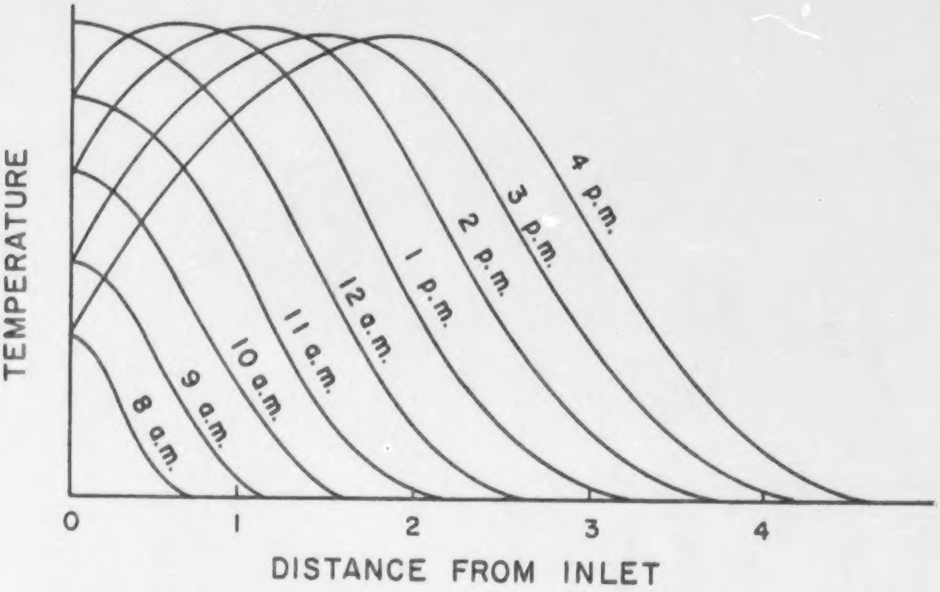


Figure 6-21: Typical Temperature Profiles in a Rock Bin Storage

Rock Size

The rock size is relatively unimportant, although it affects the pressure drop through the bed. The rock size should not be so small as to reduce flow rates significantly, nor so large that the interior of individual rocks is never heated. It is recommended that rocks from .75 to 1.5 inches in size be used.

Rounded rocks are preferable to sharply fractured rocks. However, it is entirely satisfactory to use crushed gravel aggregates normally used for concrete.

The U. S. Department of Energy has reviewed many solar installations, and has made some conclusions in the area of storage. The problems have been identified in the following excerpts.

Corrosion

Liquid solar systems are subject to corrosion and electrolytic action from metals in the various components. A drain-down system which is open to the atmosphere and continually draws in fresh air can be particularly susceptible to corrosion. As the water level fluctuates, wet metal is alternately exposed to the air and corrosion (such as rust in a steel tank) can occur. Corrosion may be inhibited by chemical additives and by the use of lined tanks.

Overheat

One potential solution to storage corrosion is the fiberglass reinforced tank. However, extreme care must be exercised in this selection, as some of the resins used by the tank manufacturer may not be rated for exposure to temperatures in excess of 160° Fahrenheit.

Certain coatings or linings can be applied to extend the operating temperature range. However, even these materials have a limit, and may cause additional problems. One storage tank developed a leak due to high temperature operation. The tank was drained and lined with a heat-resistant coating. During the next summer season, the storage temperature began to approach the limit of the new coating, and the collection system was shut-down to prevent damage to the tank. Heat buildup remained in the collectors, causing the fluid to boil out. Another storage system exceeded the limit of its tank coating; the coating melted and congealed on the inside of the collector array and piping.

Solutions to the storage overheat problem include shading of the collectors to prevent overheat or the addition of an evening heat rejection mode from storage.

Humidity

Humidity and fungal problems have been reported as a potential problem with rock storage and certain passive applications.

Of the 86 active air systems and 12 passive systems reviewed, only one site reported high humidity problems. This installation used a solar greenhouse in an indirect and isolated gain application, with two fans moving collected heat to a secondary storage area below the floor. Due to the circulation pattern of the fans, moisture transpiration from the plants raised indoor humidity, causing discomfort and some mildew during winter operation. Occupants initially corrected this condition by opening windows and turning off the fans, both increasing the heat load and decreasing the solar contribution. A better approach, later devised, was to turn off only one fan, using the other for return flow from storage and by-passing the original return flow through the building to the greenhouse. This action apparently resolved the problem.

STORAGE - EFFECTIVENESS

Size and Selection of Materials

The use of the solar system and the size of collector should govern the type and size of storage tank or rock bin selected. Materials chosen should be compatible with the range of temperatures and pressures expected during seasonal operation and periods of collector overheat or stagnation, as discussed in earlier sections.

Placement

For space heating, the ideal placement of the storage component is in the center of the conditioned space, where storage losses can indirectly reduce the heating load. However, where summer overheat of the conditioned space may be a concern, the storage component of an active system should not be located within the conditioned space, even if well insulated, and if it must be located in such a space, it should be vented to the outside.

Since passive solar storage components are normally building components, storage space is by necessity located within conditioned space. Overheat remains a concern which can be handled by means other than moving the storage, such as collector shading. Once installed, passive solar storage should not be removed or thermally degraded because the thermal capacity of the storage may be designed to prevent rapid temperature swings in the conditioned space.

One example is the placement of rugs over a thermal-storage slab floor, where a rug can prevent the solar energy from reaching the slab storage and cause room overheating.

Insulation

As discussed earlier in this module, inadequate storage insulation can be a major cause of poor system performance. Insulation should be placed on all sides of the storage container, and the insulating value should be chosed based on expected temperature of storage versus its surroundings. Several installations were well insulated. One of these used four inches of fiberglass, another used three inches of urethane. This second site, which expected a temperature-stratified condition to occur in storage, chose not to insulate the lower portion and bottom of the storage completely. Even in the lower temperature strata, the bottom portion contributed to the majority of storage losses.

Care should also be taken to protect the insulation from seepage or tank condensation, which can degrade the thermal effectiveness of the insulation.

Standby Losses

The coordination of energy use with energy availability should be a primary design objective. Many installations (mostly residential DHW) have coordinated peak demand with peak availability of solar heated water by scheduling or delaying activities until evening. Installations which do not have this potential or flexibility suffer high standby losses from the time of peak collection to the time of peak demand. One commercial site experienced its highest heat demand in the early morning, after the space heating system had been off through the night. Another school site experienced its highest domestic hot water demand in the late evening and early morning. The solar fraction for the commercial site was half the expected percentage, and the solar fraction for the DHW in the school site was only 50%, while the entire solar system was 93%. Both sites suggest that poorly timed demand probably contributed to poor water heating performance.

Temperature Stratification

Although the temperature differences within liquid storage may be small, these differences are significant to the overall functioning of the solar energy system.

Where heat enters storage, where it leaves, and where thermostatic sensors are placed is important. Failure to locate cold connections at the bottom and hot connections at the top can seriously hamper the system operation.

Stratification of storage will improve collection performance by increasing the temperature rise across the collector if the inflow is drawn from the colder strata. Stratification may decrease storage use performance if the return flow to storage enters a strata with a lower temperature than the return flow. Alternate return flow points is a method of alleviating this problem.

Any auxiliary heat elements located in the storage tank should be located in the hot portion of the storage.

Preheating Tanks

In most domestic hot water applications, potable water is preheated in a tank separate from the final hot water storage tank. This arrangement is theoretically akin to temperature stratification, and the results similar. The collectors can be operated at a greater fluid temperature change and auxiliary heat required to raise water to use temperature may not be rejected through the collectors. Systems without a preheat tank may exhibit lower performance.

When collector return temperatures are at or above the required use temperatures, a separate mode to by-pass the pre-heat mode to a direct use mode can be used.

Immersed Tanks

An immersed tank is a poor substitute for a heat exchanger. Thermal energy transfer efficiency is lower and the risk of tank corrosion is increased for non-compatible metals.

One domestic hot water installation immersed the domestic hot water tank in the main collector storage tank, to reclaim storage tank losses, to aid in preheating, and to eliminate the expense of a heat exchanger. Severe corrosion of the immersed tank occurred within only two seasons. Replacement by a compatible metal, copper coil, heat exchanger was required.

Access

Access must be provided for the repair and maintenance of storage tanks, particularly unlined steel tanks used in a drain down system. Unexpected maintenance and repairs may require inside access as well.

Air Leakage

Air leakage to or from a rock storage bin in an unconditioned space adds to the uncontrolled heat loss problem. Heat losses from storage due to air leakage have been measured as high as 50% of the stored energy. Thermal storage should be constructed of or lined with low permeability materials and sealed or otherwise fabricated to limit air leakage. The standard for sealing should be to limit leakage to 10% of system operating air flow. Construction using wood, concrete, masonry and other materials should be considered potentially porous and may require lining or sealing to limit leakage.

Fluid Leakage

Leakage of water and other storage fluids was reported at several sites, resulting in a serious degrading of tank insulation and significantly higher rates of heat loss. Leakage may result from tank corrosion or from a mismatch between tank materials and extreme system pressure and temperature levels. Fiberglass tanks, in particular, cannot withstand high temperatures or pressures, and may require special reinforcing and interior coating treatments. Wooden storage structures and tank covers are not recommended for liquids. However, kiln-dried or pressure treated lumber, with vapor barriers, have been found to be effective as outer containers. Serious design attention must be given to minimizing the risk of leakage in steel storage tanks where storage water is exposed to oxygen. One passive installation using steel water tubes experienced corrosion of the concrete slab embedded tube bottoms, which allowed the storage water to leak unnoticed below the slab.

V-S-189

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF SUBSYSTEM COMPONENTS

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

SIZING AND SELECTION OF SUBSYSTEM COMPONENTS

The collector array and storage requirements have been sized and selected in previous modules. It is now time to select and size the subsystem components - pipe/ducts, heat exchangers, pumps/blowers, valves/dampers, and associated control and protective devices.

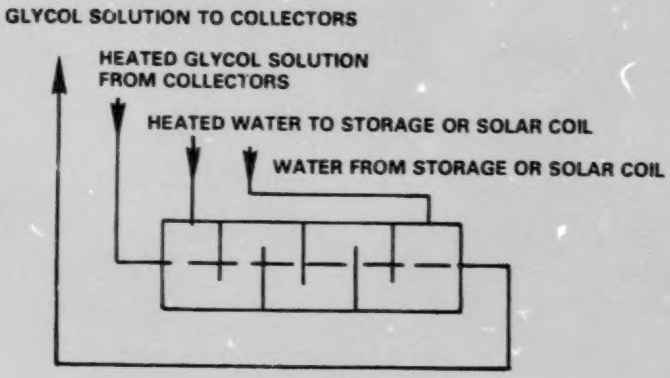
First, let's take a look at the process using a liquid system (hydronic) as an example. This method uses hydronics worksheets and the first step is selection of the heat exchanger so that the pressure drop of the exchanger can be utilized in sizing the piping and other components. Also, we will use the manufacturers' recommended sizing worksheets. The calculations in heat exchanger sizing using the thermodynamic approach are very cumbersome, so it is recommended that the student contact a professional manufacturer's representative for assistance in the selection process, if the collector manufacturer does not have a selection table such as the one used herein.

HEAT EXCHANGERS

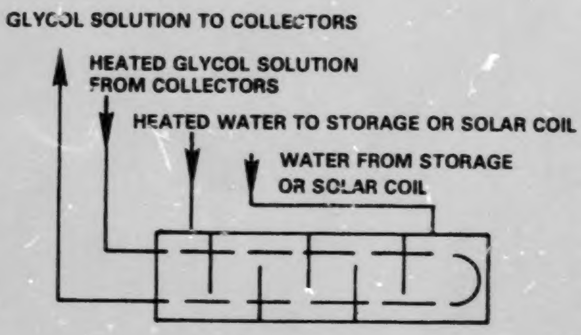
Sizing the heat exchanger can be especially complex. A table has been developed for a stated set of conditions that alleviates this complexity. The following information is intended to give the student a brief insight into some of the terms and principles involved in sizing a heat exchanger and developing the selection table.

The heat exchanger is sized according to the amount of surface area needed most to efficiently transfer collector loop heat to the remaining system loops on the opposite side of the heat exchanger. A "trade-off" exists between the solar collector area and the proper heat exchanger surface area. To explain, for the same Btu/hr. solar heat input into storage, an undersized heat exchanger will require a much larger collector surface area as compared with an adequately sized heat exchanger. Since collector surface area is more costly to attain in comparison with the heat exchanger surface area, the trade-off should be towards generous sizing of the exchanger. This is in order to maintain collection efficiency (reduced collector area for equal solar heat input into storage).

Two basic types of ethylene glycol to water heat exchanger units are described below.


GLYCOL SOLUTION TO COLLECTORS
HEATED GLYCOL SOLUTION FROM COLLECTORS
HEATED WATER TO STORAGE OR SOLAR COIL
WATER FROM STORAGE OR SOLAR COIL

COUNTERFLOW STRAIGHT TUBE HEAT EXCHANGER


GLYCOL SOLUTION TO COLLECTORS
HEATED GLYCOL SOLUTION FROM COLLECTORS
HEATED WATER TO STORAGE OR SOLAR COIL
WATER FROM STORAGE OR SOLAR COIL

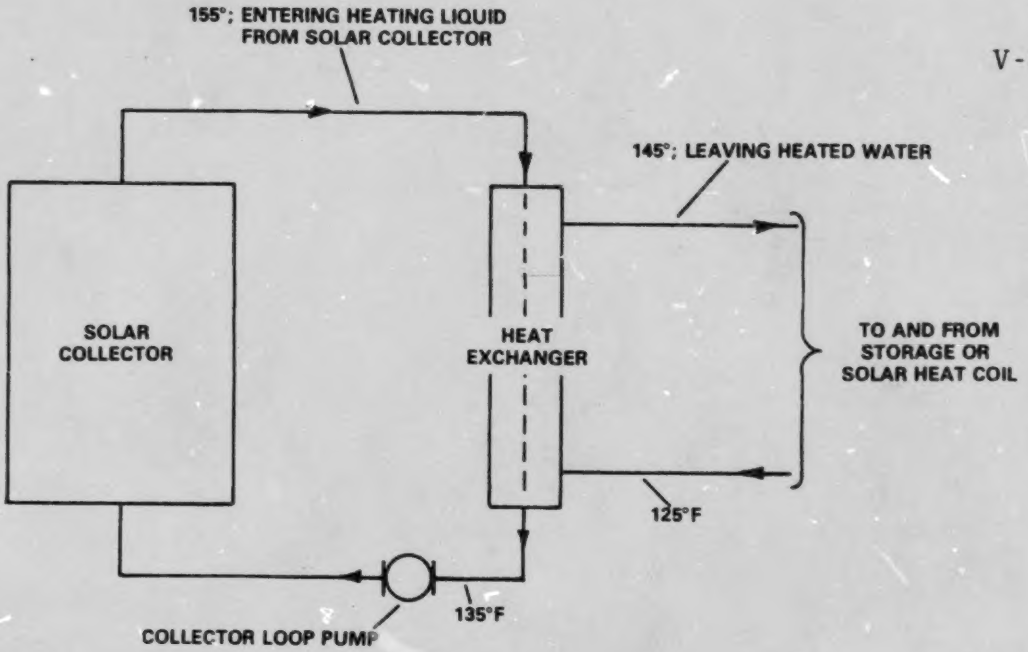
"U" TUBE HEAT EXCHANGER

The counterflow straight tube heat exchanger will be used to describe approach temperature difference, as this factor influences heat exchanger sizing and consequent collector operating efficiency.

Approach temperature difference for a heat exchanger is the difference in temperature between the incoming heating fluid and the leaving heated fluid. The following figure illustrates approach temperature difference (Δ T) for a glycol to water solar heat exchanger. The exchanger is designed for 10° "approach".

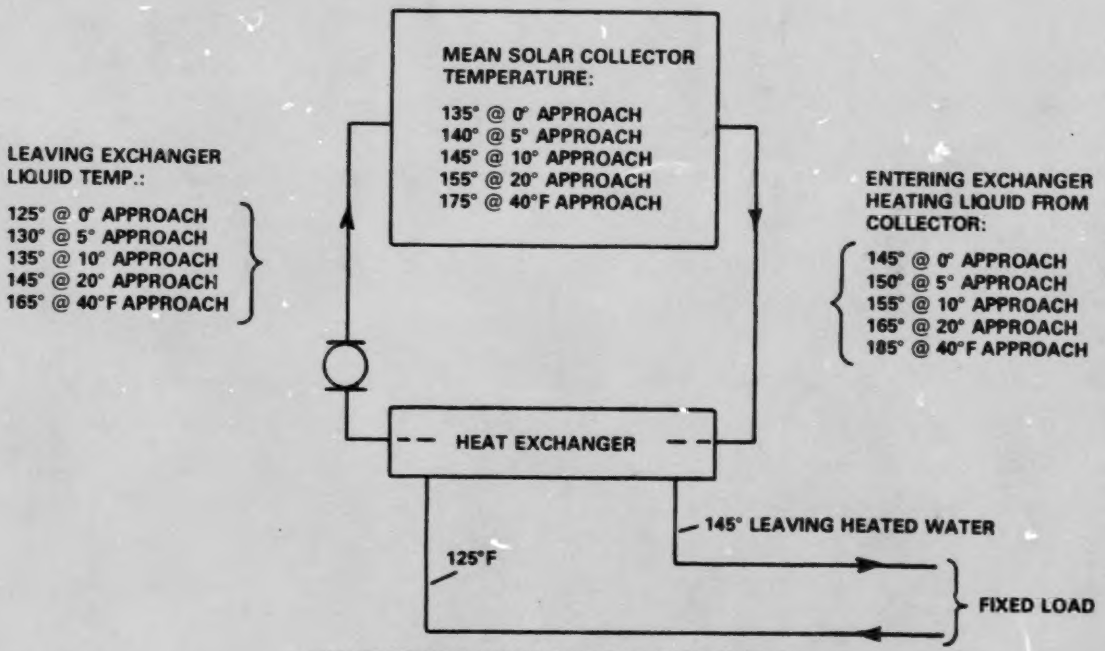
The approach Δ T used for design of the heat exchanger has great significance for glycol solution collector systems. This is because of its effect on required collector temperature to meet a stated heat exchanger load. The higher the design approach Δ T, the higher the required mean collector liquid temperature to meet the load.

The figure on the following page illustrates the effect of heat exchanger design approach Δ T on required solar collector liquid temperatures for a fixed load.


155°; ENTERING HEATING LIQUID FROM SOLAR COLLECTOR
145°; LEAVING HEATED WATER
SOLAR COLLECTOR
HEAT EXCHANGER
TO AND FROM STORAGE OR SOLAR HEAT COIL
125°F
135°F
COLLECTOR LOOP PUMP

HEAT EXCHANGER DESIGNED FOR 10° APPROACH; (155° - 145° = 10°)

The increase in mean solar panel temperature with increased heat exchanger approach temperatures will affect collection efficiency as follows for a typical collector at a 40^{o} outdoor temperature.


MEAN SOLAR COLLECTOR TEMPERATURE:
135° @ 0° APPROACH
140° @ 5° APPROACH
145° @ 10° APPROACH
155° @ 20° APPROACH
175° @ 40°F APPROACH
LEAVING EXCHANGER LIQUID TEMP.:
125° @ 0° APPROACH
130° @ 5° APPROACH
135° @ 10° APPROACH
145° @ 20° APPROACH
165° @ 40°F APPROACH
ENTERING EXCHANGER HEATING LIQUID FROM COLLECTOR:
145° @ 0° APPROACH
150° @ 5° APPROACH
155° @ 10° APPROACH
165° @ 20° APPROACH
185° @ 40°F APPROACH
HEAT EXCHANGER
145° LEAVING HEATED WATER
125°F
FIXED LOAD

CHANGE IN MEAN LIQUID TEMPERATURE IN SOLAR COLLECTORS WITH CHANGE IN HEAT EXCHANGER DESIGN APPROACH △T

Heat Exchanger Approach △T °F	Mean Collector Temp., °F	Outdoor Ambient °F	△T; Mean Collector Minus Outdoor °F	Collector Efficiency @250 B/H/FT² Insol.
0° (base)	135	40	95	35%
5°	140	40	100	33.5%
10°	145	40	105	32%
20°	155	40	115	29%
40°	175	40	135	24%

The glycol to water heat exchanger used in solar systems should be base sized for a 5 to 15° approach. The reason is that exchangers sized to less than a 5° approach will provide very little increased collector performance for a comparatively large increase in exchanger size and cost. Heat exchangers sized for over a 15° approach will reduce collector performance excessively relative to the decreased cost of the exchanger.

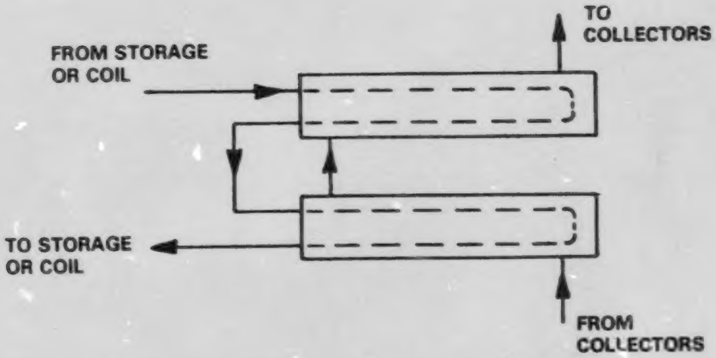
It should be noted that approach △T is a rough comparative measure of heat exchanger size. A heat exchanger selected for a 40° approach will be about 1/4 the size of an exchanger selected for a 10° approach. The difference in cost will generally not be proportionate. Considering the relatively high installed cost of the solar collectors, it would be unwise to sacrifice collector performance in order to save a little money on a smaller sized heat exchanger.

Logarithmic (Log) Mean Temperature Difference (LMTD) and Effectiveness are additional terms encountered when sizing heat exchangers. (A logarithm is a mathematical proportion normally used to shorten calculations).

The rate of heat transfer between the shell and tube of the exchanger is influenced by the temperature difference of the fluids. The greater the temperature difference, the greater the rate and amount of heat transfer. This relationship varies logarithmically, hence LMTD.

Effectiveness is the ratio of the actual rate of heat transfer between the shell and tube to the maximum rate of heat transfer between the shell and tube if the tube bundle had an infinite surface area. A heat exchanger for a solar system should have an Effectiveness of 0.6 or greater.

Sometimes, it is more economical to achieve adequate Effectiveness by employing two small heat exchangers rather than one bigger one. This is the case in the Ames example system. Shown below is a diagram of two exchangers piped in a series-counterflow configuration.


TO COLLECTORS
FROM STORAGE OR COIL
TO STORAGE OR COIL
FROM COLLECTORS

SERIES-COUNTERFLOW CONFIGURATION

STEP 3A. HEAT EXCHANGER SIZING WORKSHEET

CAUTION: The following table is based on the assumptions listed below and is not intended to cover all field applications. For systems that vary from the stated assumptions, contact the ITT Bell & Gossett representative in your area for heat exchanger verification.

HEAT EXCHANGER SELECTION TABLE

Number of Collectors	Heat Exchanger*		Flow Rate (gpm)		Velocity (fps)		Friction Loss (ft. of head)		Shell Diameter (in.)	Shell Length (in.)
	Quantity	Model No.	Shell	Tube	Shell	Tube	Shell	Tube		
4	2	STH-310-4	1.84	1.64	.60	1.22	.9	1.52	3	17
5	2	STH-315-4	2.3	2.04	.74	1.51	1.14	2.36	3	23
9	"	"	4.14	3.68	1.33	2.73	4.3	6.52	"	"
10	2	STH-320-4	4.6	4.08	1.48	3.02	6.8	8.16	3	29
11	"	"	5.06	4.48	1.63	3.32	8.16	10.16	"	"
12	2	STH-415-4	5.52	4.9	1.05	2.11	2.94	2.68	4	23
14	"	"	6.4	5.73	1.22	2.47	3.86	3.16	"	"
15	2	STH-420-4	6.9	6.12	1.32	2.64	5.68	4.26	4	29
18	"	"	8.3	7.3	1.58	3.14	7.94	5.46	"	"
20	"	"	9.2	8.16	1.75	3.5	9.98	7.3	"	"
21	2	STH-520-4	9.66	8.56	1.26	1.68	2.94	2.4	5	30
22	2	STH-530-4	10.12	8.99	1.32	1.77	4.76	2.94	5	42
25	"	"	11.5	10.2	1.5	2.01	5.9	3.84	"	"
30	"	"	13.8	12.25	1.8	2.41	7.48	5.4	"	"
35	"	"	16.1	14.3	2.09	2.81	9.3	6.54	"	"
37	"	"	17.02	15.11	2.22	2.97	12.92	7.1	"	"
38	2	STH-620-4	17.48	15.5	1.56	1.85	4.76	1.9	6	30
39	2	STH-630-4	17.94	15.93	1.6	1.9	6.58	2.7	6	42
40	"	"	18.4	16.32	1.64	1.95	7.72	2.76	"	"
72	"	"	33.12	29.4	2.96	3.51	22.68	5.94	"	"

**Table is based on the following assumptions: 1) The heat exchangers in this table are ITT Bell & Gossett models with a 10° approach; 2) Log mean temperature difference (LMTD) between shell and tube = 10°; 3) Heat exchangers are manifolded in series-counterflow configuration; 4) Collector flow rate is .4 gpm; 5) Temperature drop is the same for collector and Load side loops; 6) Load side flow rate to collector flow rate ratio is .88696; and 7) Solar Incident Radiation is 220 Btuh/sq. ft.*

In the model number STH-315-4, the 3 signifies a 3 inch diameter shell; the 15 signifies a 15 inch tube length; and the 4 a 4 pass heat exchanger. (A 4 pass exchanger means each tube winds the length of the shell four times.)

Heat Exchanger Selected: STH-420-4 (2, PIPED IN SERIES-COUNTERFLOW)

SHELL DIAMETER	VOLUMES IN GALLONS PER LINEAR INCH	
	IN SHELL	IN TUBES
4	.036	.019
6	.083	.042
8	.15	.075
10	.2	.1
12	.333	.183
14	.417	.217
16	.542	.291
18	.666	.375
20	.833	.459
24	1.25	.625

AVERAGE WATER VOLUME FOR ITT BELL & GOSSETT SHELL AND TUBE HEAT EXCHANGER

Now that a background in heat exchanger selection has been established, we can proceed to pipe sizing. A brief discussion of hydronic principle will be helpful at this time. Also, keep in mind that the first, initial pipe size is tentative and may have to be changed as a result of pump selection.

HYDRONIC COMPONENTS

Prior to beginning the hydronic selection process, detailed drawings of the entire system must be made. The designer may want to make a schematic drawing first. This schematic should show all of the pieces and parts of the system and their approximate layout. Such a drawing is helpful prior to making up a bill of materials. A list of schematic symbols and a schematic of the Ames example system follow in this section.

A more refined pictorial drawing should also be made. The exact system layout should be represented, including dimensions between components (this must be known to select certain hydronic components accurately). This type drawing of the Ames example system follows in this section. A cutaway of the house graphically shows some of the considerations that have to be made when planning the system layout. Note that the collector array is piped in a two row, reverse return flow configuration.

Hydronic components are sized according to the fluid flow demands of each loop. In the typical solar system there are two basic flow demands, one for the collector loop and one for the load side loops. Load side loops consist of the heat exchanger to solar coil loop, heat exchanger to storage loop and storage to solar coil loop.

There are four basic steps to sizing solar hydronic components. Stem 1 determines fluid flow demands; Step 2 sizes pipe and determines friction losses; Step 3 balances fluid flow, and Step 4 sizes the pump. In addition, the collector loop has Step 5, which sizes the compression tank. It is important to note that the hydronics industry (unlike the refrigeration industry) measures pipe size diameter in nominal inches, which is the outside diameter dimension. Also, copper tubing will carry the fluid in most typical solar systems.

PRESSURE DROP OR FRICTION LOSS

Pressure drop (sometimes called friction loss) is the term meaning power is consumed in moving fluids through such things as pipes, fittings and heating units. Expressed in another way, pressure drop is the amount of pressure lost between any two points in a system. For example, if the city water pressure at the inlet of a copper coil is 40 pounds per square inch (psi), and at the outlet 35 psi there is a 5 psi pressure drop through the heater.

Manufacturers who publish pressure drop information on their equipment may express the date either in pounds per square inch, in feet of water, (sometimes called feet of heat) or milinches.

These figures are easily interchangeable as follows:

1 psi = 2.3 feet of water
1 foot of water = .43 psi
1 foot of water = 12,000 milinches
Therefore, 1 inch of water = 1000 milinches.

Pressure drop is caused by the friction created between the inner walls of the conveyer and the moving liquid. In a horizontal pipe in which there is no flow, the pressure is equal at all points. The moment flow starts friction is set up, which increases in direct proportion to the velocity of the flow.

To calculate the change in pressure drop when you have an increase or decrease in flow (expressed in gallons per minute or gpm), this simple rule may be followed: "Divide final gpm by initial gpm and square result. Multiply this result by initial pressure drop. This result is the new pressure drop."

The following example shows the effect of increasing the gpm from 3 to 6 in a system with an initial pressure drop of 5 lbs.

$$\left(\frac{\text{Final gpm}}{\text{Initial gpm}}^{2}\right) \times \text{Original Pressure Drop} = \text{New Pressure Drop}$$

$$\frac{6}{3} = 2 \quad 2^2 = 4 \quad 4 \times 5 = 20 \text{ psi.}$$

Velocity in feet per second (fps) may be substituted for gpm in the previous formula.

Therefore, in designing both service water heating systems and hot water space heating systems, pressure drop must be taken into consideration. In each case, enough power must be available to overcome the effects of pressure drop, before the desired results can be obtained. This means that the power consumption, or pressure drop, of each component part of a system must be known, and a source of sufficient power provided. In a forced hot water heating system this power is provided by the pump - in a domestic water heating system, city water pressure is the source of power.

HEAD PRESSURE

Head pressure is used in designating the capacity of a circulating pump, and is merely another way of expressing pressure drop. The maximum head of a pump is actually the maximum pressure drop, against which the pump can induce a flow of liquid. Head pressure is usually expressed in feet of water, or feet of head.

STATIC PRESSURE

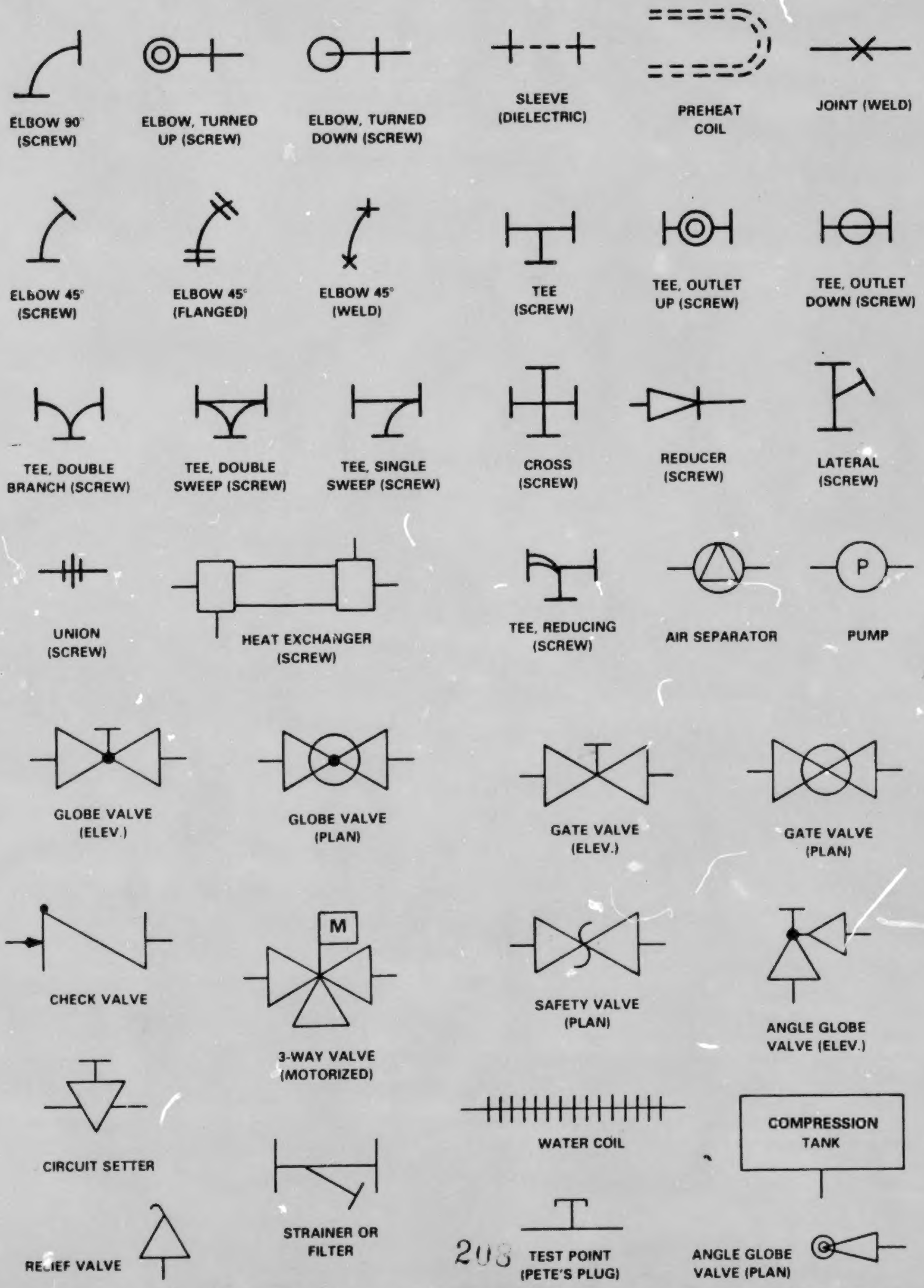
Head pressure should not be confused with static pressure, as they have no relationship. Static pressure is created by the weight of fluid in the system.

Static pressure has no effect on pump capacity in pressurized or closed liquid systems (most solar systems are pressurized). If you will consider a hot water heating system as being an upright loop of water confined in a pipe, the static pressure in one of the vertical pipes of the loop is identical with the pressure at the same level in the opposite vertical pipe.

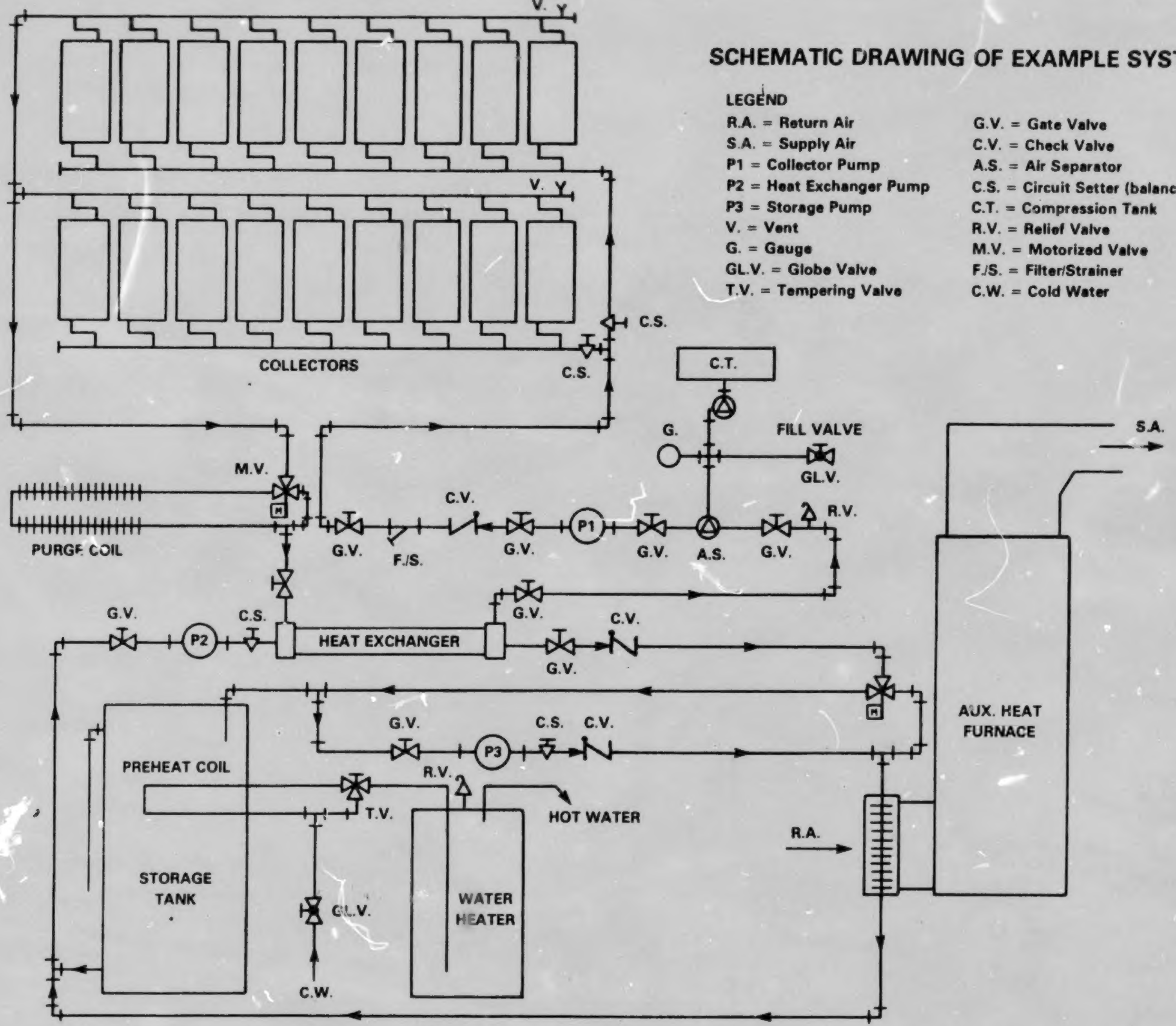
The static pressure at the point where the pump is installed in a closed system is exactly equalized by the pressure at the same level in the opposite side of the loop. The capacity of the pump, then, is limited only by the friction or head pressure in the system.

In liquid non-pressurized or "open" systems, static pressure must be considered in addition to head pressure in order to size the pump. The static pressure is equal to .43 psi per foot of height above the pump. For example, assume the highest pipe run is 20 feet above the pump. The static pressure at the pump will be 20 times .43, which equals 8.6 psi. At various elevations above the pump, the static pressure becomes correspondingly less. At 10 feet, it is 4.3 psi. At the top pipe run, located 20 feet above the pump, there is no static pressure.

SCHEMATIC SYMBOLS FOR HYDRONIC COMPONENTS


ELBOW 90°
(SCREW)
ELBOW, TURNED
UP (SCREW)
ELBOW, TURNED
DOWN (SCREW)
SLEEVE
(DIELECTRIC)
PREHEAT
COIL
JOINT (WELD)
ELBOW 45°
(SCREW)
ELBOW 45°
(FLANGED)
ELBOW 45°
(WELD)
TEE
(SCREW)
TEE, OUTLET
UP (SCREW)
TEE, OUTLET
DOWN (SCREW)
TEE, DOUBLE
BRANCH (SCREW)
TEE, DOUBLE
SWEEP (SCREW)
TEE, SINGLE
SWEEP (SCREW)
CROSS
(SCREW)
REDUCER
(SCREW)
LATERAL
(SCREW)
UNION
(SCREW)
HEAT EXCHANGER
(SCREW)
TEE, REDUCING
(SCREW)
AIR SEPARATOR
P
PUMP
GLOBE VALVE
(ELEV.)
GLOBE VALVE
(PLAN)
GATE VALVE
(ELEV.)
GATE VALVE
(PLAN)
CHECK VALVE
M
3-WAY VALVE
(MOTORIZED)
SAFETY VALVE
(PLAN)
ANGLE GLOBE
VALVE (ELEV.)
CIRCUIT SETTER
WATER COIL
COMPRESSION
TANK
RELIEF VALVE
STRAINER OR
FILTER
TEST POINT
(PETE'S PLUG)
ANGLE GLOBE
VALVE (PLAN)

SCHEMATIC DRAWING OF EXAMPLE SYSTEM


V.
V.
COLLECTORS
C.S.
C.S.
C.T.
G.
FILL VALVE
GL.V.
S.A.
M.V.
PURGE COIL
G.V.
F./S.
C.V.
G.V.
P1
G.V.
A.S.
G.V.
R.V.
G.V.
G.V.
P2
C.S.
HEAT EXCHANGER
G.V.
C.V.
G.V.
C.S.
C.V.
P3
PREHEAT COIL
R.V.
T.V.
HOT WATER
R.A.
AUX. HEAT FURNACE
STORAGE TANK
GL.V.
WATER HEATER
C.W.

LEGEND

R.A. = Return Air
S.A. = Supply Air
P1 = Collector Pump
P2 = Heat Exchanger Pump
P3 = Storage Pump
V. = Vent
G. = Gauge
GL.V. = Globe Valve
T.V. = Tempering Valve
G.V. = Gate Valve
C.V. = Check Valve
A.S. = Air Separator
C.S. = Circuit Setter (balance valve)
C.T. = Compression Tank
R.V. = Relief Valve
M.V. = Motorized Valve
F./S. = Filter/Strainer
C.W. = Cold Water

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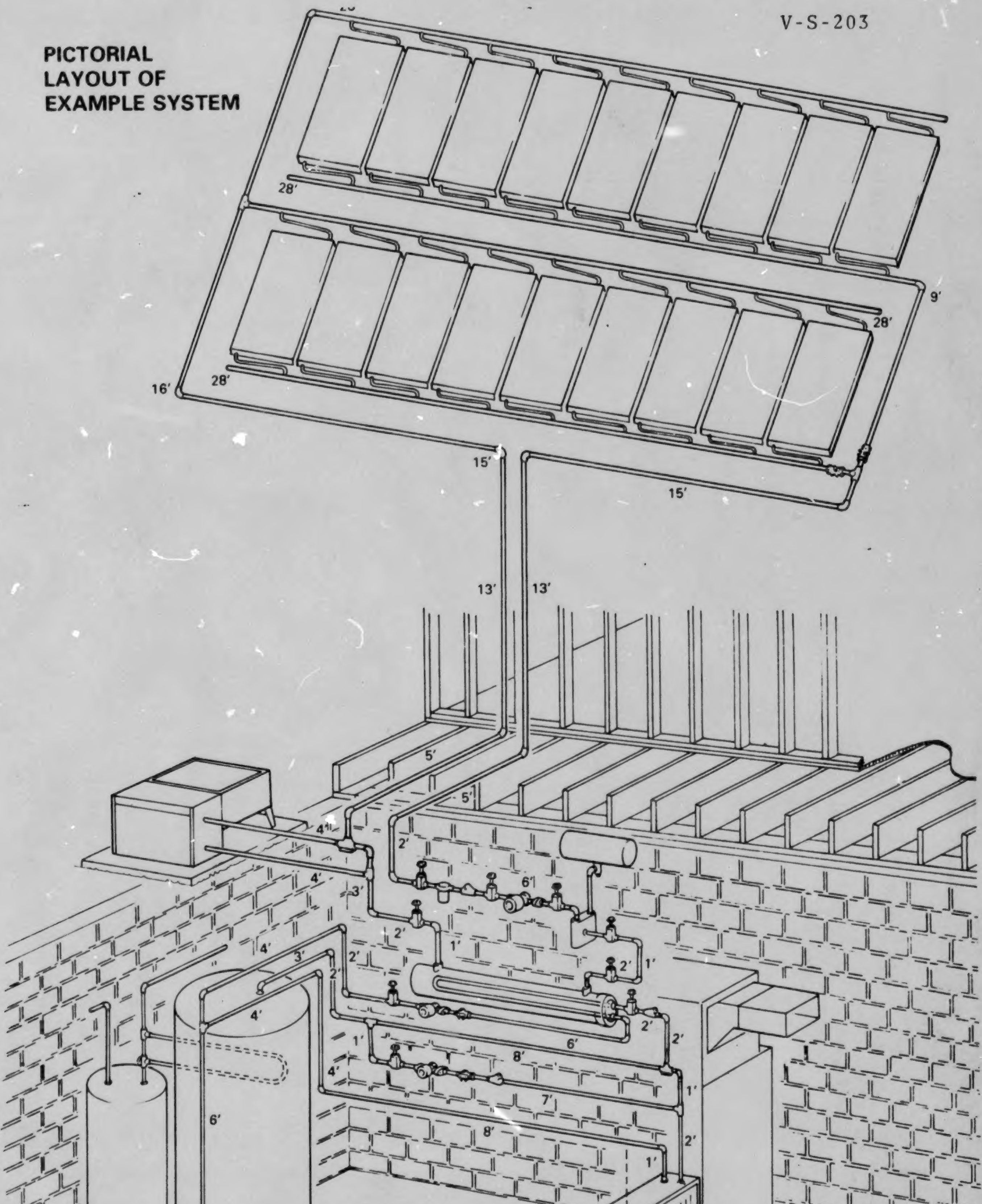
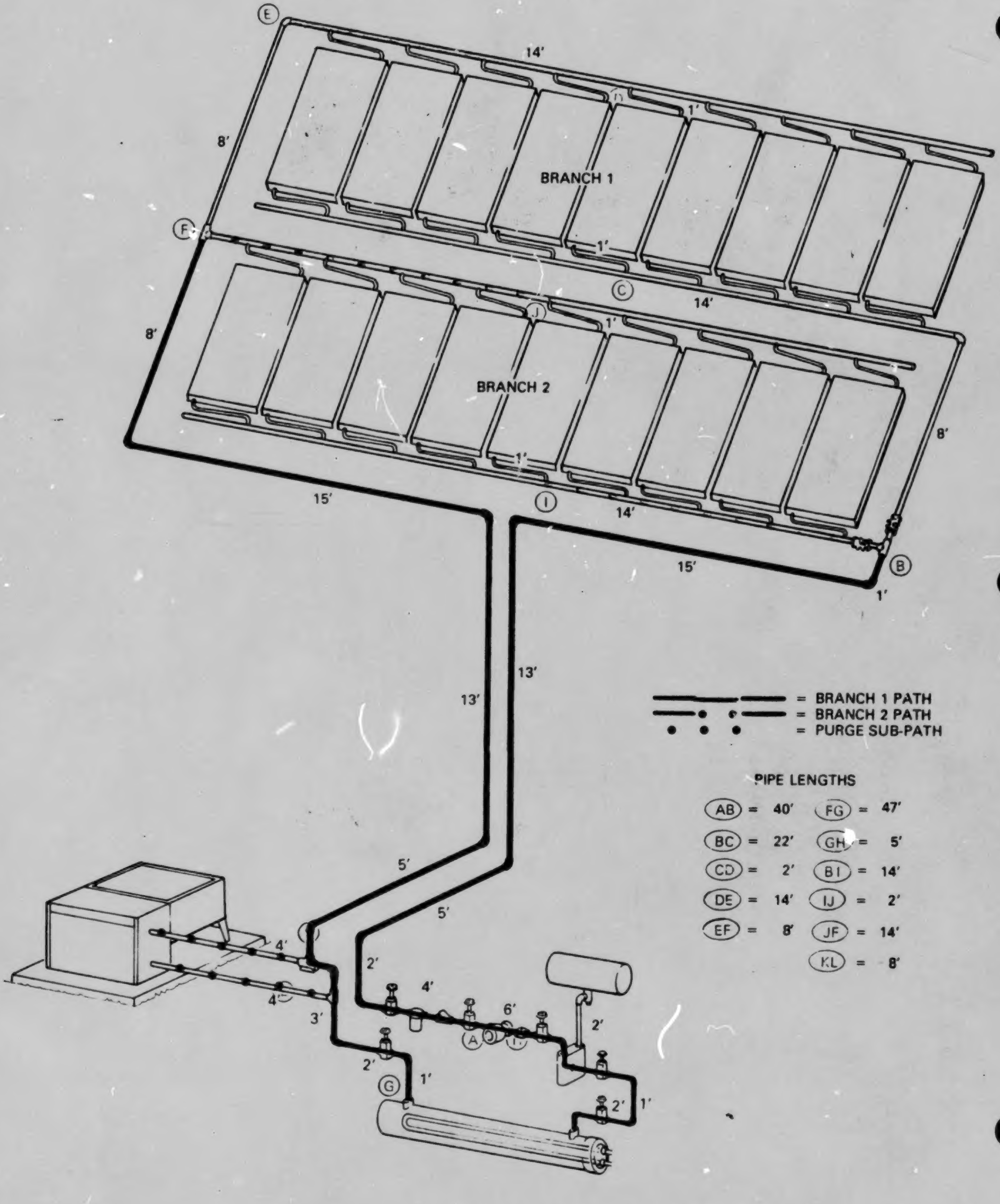
PICTORIAL LAYOUT OF EXAMPLE SYSTEM


28'
28'
9'
28'
16'
15'
15'
13'
13'
5'
5'
4'
4'
3'
2'
6'
2'
1'
2'
1'
4'
3'
2'
2'
4'
2'
2'
1'
6'
8'
4'
7'
1'
6'
8'
2'
1'

COLLECTOR LOOP SIZING


E
14'
8'
1'
BRANCH 1
F
1'
C
14'
J
1'
8'
BRANCH 2
8'
1'
15'
I
14'
15'
B
1'
13'
13'
5'
5'
4'
4'
3'
2'
2'
4'
6'
A
2'
G
1'
2'
1'
= BRANCH 1 PATH
= BRANCH 2 PATH
● ● ● = PURGE SUB-PATH

PIPE LENGTHS

AB = 40'
BC = 22'
CD = 2'
DE = 14'
EF = 8'
FG = 47'
GH = 5'
BI = 14'
IJ = 2'
JF = 14'
KL = 8'

COLLECTOR LOOP

FLUID FLOW WORKSHEET

Line 1. Number Of Collectors:

a. Branch 1 (from System Layout Drawing) .... 9 ✓

b. Branch 2 (from System Layout Drawing) 9

c. Branch 3 (from System Layout Drawing) .... —

d. TOTAL Number Of Collectors In System 18

Line 2. Collector Flow Rate:

a. Recommended Flow Rate Through Collector (from Collector Engineering Handbook sheet in DATA section .... 0.4 gpm

b. Correction factor for Ethylene Glycol (from Table 3a of DATA section) 1.15

NOTE: Average temperature of collector loop fluid in most typical systems is approximately 140°F. Enter Table 3a at 140°.

c. Corrected Design Flow Rate:
(Line 2a 0.4) x (Line 2b 1.15) = ... 0.46 gpm

Line 3. Flow Rates:

a. Branch 1 Flow Rate: (Line 1a 9) x (Line 2c 0.46) = 4.14 OR 4.2 gpm

b. Branch 2 Flow Rate: (Line 1b 9) x (Line 2c 0.46) = 4.14 OR 4.2 gpm

c. Branch 3 Flow Rate: (Line 1c —) x (Line 2c —) = — gpm

d. TOTAL Loop Flow Rate .... 8.4 gpm

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MASTER HYDRONICS WORKSHEET

FOR COLLECTOR LOOP – BRANCH 1 PATH
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION AB							
PIPE 1	8.4	1¼	2.1	40	40.0	40.0	0.80
GATE VALVES 2	"	"	"	—	1.5	3.0	.06
CHECK VALVE (SWING) 1	"	"	"	—	14.0	14.0	.30
ELBOWS (ELS) (90° LONG RAD) 5	"	"	"	—	2.3	11.5	.22
TEE (STRAIGHT FLOW) 1	"	"	"	—	2.3	2.3	.05
STRAINER 1	(FROM TABLE 6B) →						.23
							1.66
SECTION BC							
PIPE 1	4.2	1	1.6	22	22.0	22.0	.35
SUDDEN CONTRACTION (.8=¾ 1	"	1¼ → 1	"	—	.7	.7	.01
EL (90° L.R.) 1	"	1	"	—	1.7	1.7	.03
TEE (BRANCH FLOW) 1	"	1	"	—	5.0	5.0	.10
NOTE: BALANCE VALVE FRICTION LOSS & SETTING FIGURED IN STEP 4C.							.49
SECTION CD							
PIPE (FLEX HOSE) 1	.46	3/8	1.0	2	2.0	2.0	.04
SUDDEN CONTR. (.375=¼ 1	"	1 → 3/8	"	—	.7	.7	.01
ELS OFF COLLECTOR (90° STD) 2	"	3/8	"	—	1.4	2.8	.06
TEE (BRANCH FLOW) 1	"	3/8	"	—	2.7	2.7	.05
COLLECTOR 1	(FROM COLLECTOR EHB DATA) →						.45
							.61

MASTER HYDRONICS WORKSHEET

FOR COLLECTOR LOOP-- BRANCH 1 PATH (CONT.)
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (In., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION DE							
PIPE 1	4.2	1	1.6	14	14.0	14.0	0.23
SUDDEN ENLARGEMENT (1/4) 1	"	3/8 → 1	"	—	1.4	1.4	.03
EL (90° L.R.) 1	"	1	"	—	1.7	1.7	.02
							.28
SECTION EF							
DOWNRUNNER PIPE 1	4.2	3/4	2.8	8	8.0	8.0	.45
SUDDEN CONTR. (3/4) 1	"	1 → 3/4	"	—	.5	.5	.06
TEE (STRAIGHT FLOW) 1	"	3/4	"	—	1.4	1.4	.10
							.61
SECTION FG							
DOWNRUNNER PIPE 1	8.4	1 1/4	2.1	47	47.0	47.0	.90
SUDDEN ENLARGEMENT (1/2) 1	"	3/4 → 1 1/4	"	—	1.5	1.5	.03
ELS (90° L.R.) 7	"	1 1/4	"	—	2.3	16.1	.30
TEE (STRAIGHT FLOW) 1	"	"	"	—	2.3	2.3	.05
GATE VALVE 1	"	"	"	—	1.5	1.5	.03
3-WAY VALVE (DIVERTING) 1		(CV=17, FROM TABLE 6c)					.65
							1.96

TOTAL 2.85

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MASTER HYDRONICS WORKSHEET

FOR COLLECTOR LOOP-- BRANCH 1 PATH (CONT.)
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A **SECTION & COMPONENT** (from System Layout Drawing)	**Col. B** **FLUID FLOW (gpm)** (from Line 3 of Step 4A Worksheet)	**Col. C** **PIPE SIZE (in., I.D.)** (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	**Col. D** **VELOCITY (fps)** (from Table 4 of DATA section)	**Col. E** **PIPE LENGTH (ft.)** (from System Layout Drawing)	**Col. F** **EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.)** (from Tables 5a, 5b or 5c of DATA Section)	**Col. G** **TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.):** (Col. A) x (Col. F)	**Col. H.** **FRICTION LOSS (ft of head)** (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION G H							
1 Pipe	8.4	1¼	2.1	5	5.0	5.0	.10
4 ELS (90° L.R.)	"	"	"	—	2.3	9.2	.20
3 GATE VALUES	"	"	"	—	1.5	4.5	.09
1 AIR SEPARATOR (SHARP EDGE)	"	"	"	—	ENTRANCE: 5.3 EXIT : 2.6	5.3+2.6= 7.9	.20
1 HEAT EXCHANGER		(FROM STEP 3A WORKSHEET)			→	→	7.94
							(8.53)

MASTER HYDRONICS WORKSHEET

FOR COLLECTOR LOOP--BRANCH Z PATH

USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION AB			(FIGURED PREVIOUSLY)			→	1.66
SECTION B1							
PIPE 1	4.2	1	1.6	14	14.0	14.0	.20
SUDDEN CONTR. (3/4) 1	"	1¼→1	"	—	.7	.7	.01
TEE (BRANCH FLOW) 1	"	1	"	—	5.0	5.0	.07
NOTE:	BALANCE VALVE FRICTION LOSS & SETTING FIGURED IN STEP 4C						.28
SECTION 1J							
PIPE (FLEX HOSE) 1	.46	3/8	1.0	2	2.0	2.0	.04
SUDDEN CONTR. (1/4) 1	"	1→3/8	"	—	.7	.7	.01
ELS OFF COLLECTOR/90°STD 2	"	3/8	"	—	1.4	2.8	.06
TEE (BRANCH FLOW) 1	"	"	"	—	2.7	2.7	.05
COLLECTOR 1		(FROM COLLECTOR EHB DATA)				→	.45
							.61
SECTION JF							
PIPE 1	4.2	1	1.6	14	14.0	14.0	.20
SUDDEN ENLARGEMENT (1/4) 1	"	3/8→1	"	—	1.4	1.4	.02
TEE (BRANCH FLOW) 1	"	1	"	—	5.0	5.0	.07
							.29
SECTION FG			(FIGURED PREVIOUSLY)			→	1.96
SECTION GH			(FIGURED PREVIOUSLY)			→	8.53

TOTAL 13.33

MASTER HYDRONICS WORKSHEET

FOR COLLECTOR LOOP -- PURGE SUB-PATH
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION K1.							
PIPE 1	8.4	1 1/4	2.1	8	8.0	8.0	0.16
TEE (BRANCH FLOW) 1	"	"	"	—	7.0	7.0	.14
PURGE COIL 1	(FROM STEP 3B WORKSHEET) →						3.02
							3.32

COLLECTOR LOOP

FLOW BALANCE WORKSHEET

Line 1. Friction Loss Of Path(s) (from Step 4B Worksheets)*:
a. Branch 1 Path .... 14.14 ft. of head
b. Branch 2 Path .... 13.33 ft. of head
c. Branch 3 Path .... — ft. of head

Line 2. Friction Loss Correction Factor for Ethylene Glycol (from Table 3b of DATA section) 1.13
NOTE: Average temperature of collector loop fluid in most typical systems is approximately 140°F. Enter Table 3b at 140°.

Line 3. Friction Losses Corrected For Ethylene Glycol:
a. (Path 1 14.14) x (Line 2 1.13) = ... 15.98 ft. of head
b. (Path 2 13.33) x (Line 2 1.13) = ... 15.06 ft. of head
c. (Path 3 —) x (Line 2 —) = ... — ft. of head
d. Path With Greatest Friction Loss (called the Critical Path) = Path 1

Line 4. Determine Presetting Of Critical Path Balance Valve:
a. (Line 3d Critical Path Friction Loss 15.98) x 20% = ... 3.20 ft. of head
b. Use Circuit Setter Calculator Wheel — Preset Side To Determine Setting NeededTo Produce Line 4a Friction Loss: .... 1"B.V. 15 ° closed

NOTE: This practice acts as an insurance policy against hydronic sizing miscalculations. Partial closing of this valve in the preset stage allows flow to be adjusted in either direction (increased as well as decreased) in the actual balancing procedure. A circuit setter wheel is in the back pocket of this binder.

It is recommended that a balance valve be installed even if the loop has no branches and should theoretically need no balancing. The balance valve provides a convenient means of adjusting flow for whatever the reason.

Line 5. Design Friction Loss Of Loop:
(Line 3 Critical Path Friction Loss 15.98) + (Line 4a 3.20) = ... 19.18 ft. of head

Line 6. Determine Presetting Of Balance Valve(s) In Non-Critical Path(s):
a. (Line 5 19.18) – (Non-Critical Path Friction Loss From Line 3 15.06) = 4.12 ft. of head
b. Use Circuit Setter Calculator — Preset Side To Determine Setting Needed To Produce Line 6a Friction Loss: .... 1"B.V. 18 ° closed
c. (Line 5 —) – (Non-Critical Path Friction Loss From Line 3 —) = — ft. of head
d. Use Circuit Setter Calculator — Preset Side To Determine Setting Needed To Produce Line 6a Friction Loss: .... — ° closed

NOTE: This action balances or equalizes the loop rates throughout the loop.

*Purge sub-path is not to be considered in the balancing process.

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COLLECTOR LOOP

PUMP SELECTION WORKSHEET

Collector Loop Not In Purge Cycle

Line 1. a. Design Friction Loss (from Line 5 of Step 4C Worksheet) .... 19.18 ft. of head

b. Design Flow Rate (from Line 3d of Step 4A Worksheet) .... 8.4 gpm

NOTE: The point on a graph formed by the intersection of the design friction loss and design flow rate is called the system design point. This is the focal point for properly sizing a pump.

Line 2. Select Pump Performance Curve graph (see Tables 7a or 7b in DATA section) whose range is such that the system design point falls near mid-range on the graph.

Line 3. Calculate at least four points of System Performance Curve from the following formula:

$$\left(\frac{Q2}{Q1}\right)^2 = \frac{h2}{h1}$$

Where:

Q2 = final flow (arbitrary number)

Q1 = known flow (from Line 1b)

h2 = final friction loss (this is to be determined)

h1 = known friction loss (from Line 1a)

$\left(\frac{3}{8.4}\right)^2 = \frac{h2}{19.18}$ h2 = 2.45

$\left(\frac{5}{8.4}\right)^2 = \frac{h2}{19.18}$ h2 = 6.80

$\left(\frac{10}{8.4}\right)^2 = \frac{h2}{19.18}$ h2 = 27.18

$\left(\frac{12}{8.4}\right)^2 = \frac{h2}{19.18}$ h2 = 39.14

NOTE: The system curve shows how a change in flow affects friction loss, and vice versa. System curve analysis identifies possible system operating characteristics and helps prevent pump sizing mistakes.

Line 4. Plot the system design point and the four (Q2, H2 System Curve) points calculated in Line 3 on Pump Performance Curve graph. (See next page.)

Collector Loop In Purge Cycle

Line 5. Friction Loss Of Purge Sub-Path Corrected For Ethylene Glycol:
(Purge Friction Loss from Step 4B Worksheet 3.32) x
(Correction Factor from Table 3b of DATA section 1.13) = ... 3.75 ft. of head

Line 6. a. Total Friction Loss Of Collector Loop In Purge Cycle:
(Line 1a 19.18) + (Line 5 3.75) = ... 22.93 ft. of head

b. Design Flow Rate (from Line 3d of Step 4A Worksheet) .... 8.4 gpm

NOTE: The intersection of the above numbers on a graph is the system design point.

Line 7. Calculate at least four points of System Performance Curve In Purge Cycle (use formula from Line 3 with one exception; h1 figure is from Line 6a instead of Line 1a):

$\left(\frac{3}{8.4}\right)^2 = \frac{h2}{22.93}$ h2 = 2.92

$\left(\frac{5}{8.4}\right)^2 = \frac{h2}{22.93}$ h2 = 8.12

$\left(\frac{10}{8.4}\right)^2 = \frac{h2}{22.93}$ h2 = 32.50

$\left(\frac{12}{8.4}\right)^2 = \frac{h2}{22.93}$ h2 = 46.80

Line 8. Plot the system design point and the four (Q2, h2) points calculated in Line 7 on Pump Performance Curve graph. (See next page.)

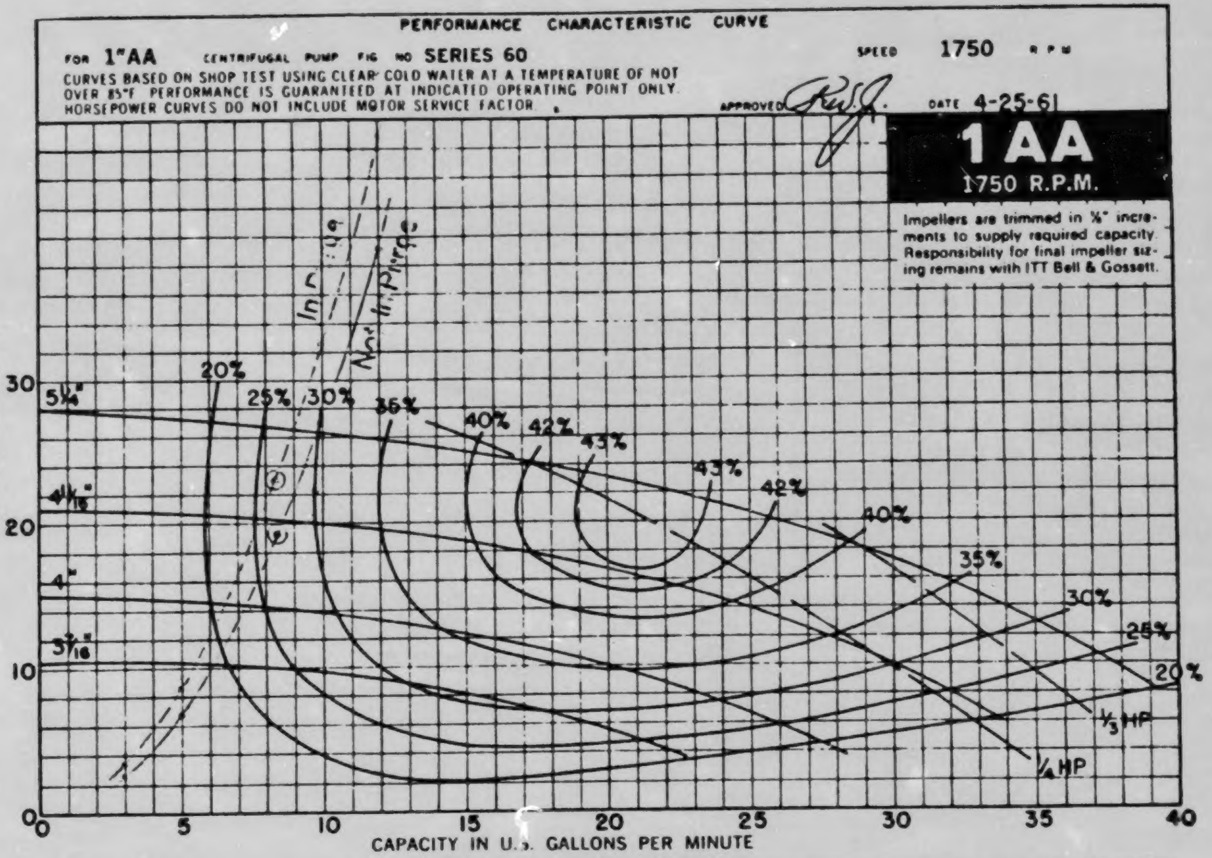
Pump Selection

Line 9. Select pump whose performance curve is nearest the system design point. If the design point is equidistant between two pump curves, normally size to the higher pump curve. The system design point should also intersect the pump curve in the middle one-third range of pump curve, or slightly left of middle.

Pump Selected: rpm 1750 hp 1/4 impeller 4 11/16 model no. 1AA, SERIES 60

NOTE: Since system will be operating "not in purge" more than "in purge", the pump normally should be sized to the "not in purge" design point. Check "in purge" system curve to see that the reduced gpm output of the pump in purge will not significantly affect system operation

PUMP PERFORMANCE CURVES — ITT BELL & GOSSETT


PERFORMANCE CHARACTERISTIC CURVE

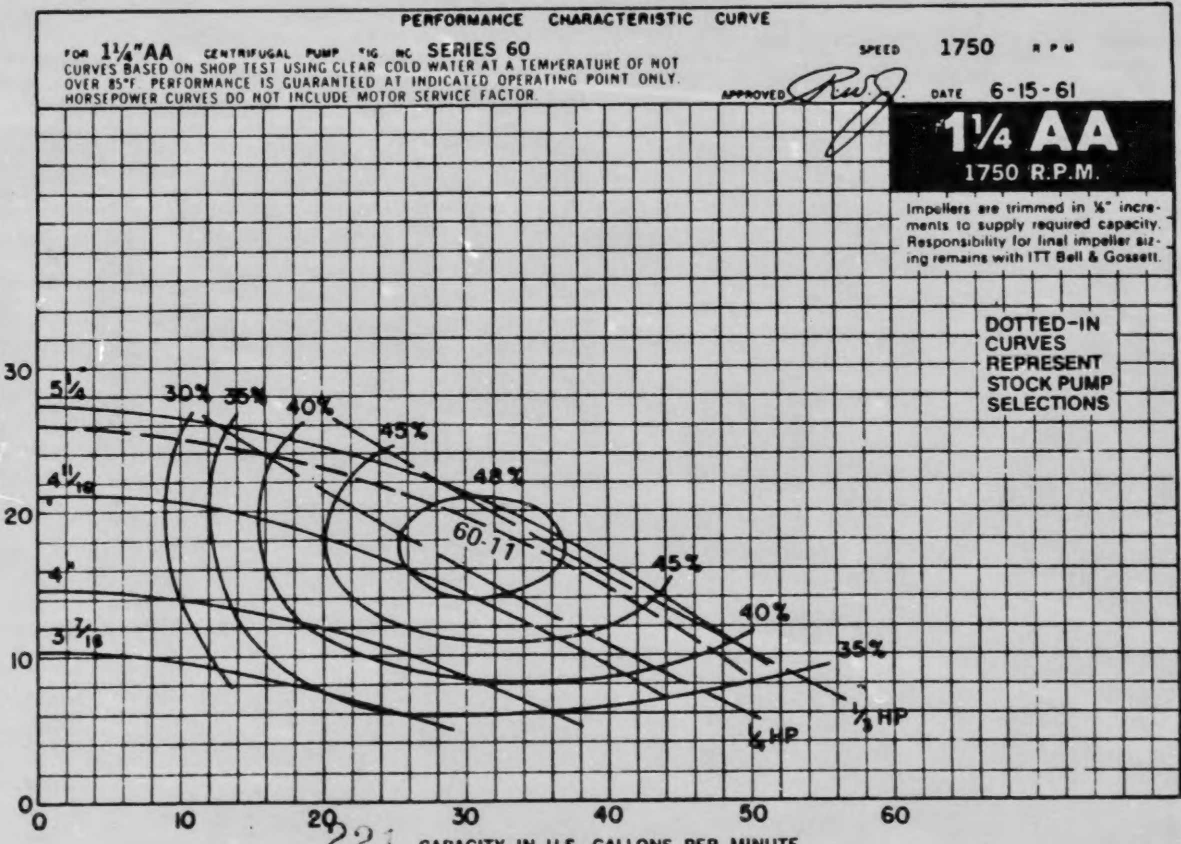
FOR 1"AA CENTRIFUGAL PUMP FIG NO SERIES 60 SPEED 1750 RPM

CURVES BASED ON SHOP TEST USING CLEAR COLD WATER AT A TEMPERATURE OF NOT OVER 85°F. PERFORMANCE IS GUARANTEED AT INDICATED OPERATING POINT ONLY. HORSEPOWER CURVES DO NOT INCLUDE MOTOR SERVICE FACTOR.

APPROVED RWJ DATE 4-25-61

1 AA
1750 R.P.M.

Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett.


PERFORMANCE CHARACTERISTIC CURVE

FOR 1¼"AA CENTRIFUGAL PUMP FIG NO SERIES 60 SPEED 1750 RPM

CURVES BASED ON SHOP TEST USING CLEAR COLD WATER AT A TEMPERATURE OF NOT OVER 85°F. PERFORMANCE IS GUARANTEED AT INDICATED OPERATING POINT ONLY. HORSEPOWER CURVES DO NOT INCLUDE MOTOR SERVICE FACTOR.

APPROVED RWJ DATE 6-15-61

1¼ AA
1750 R.P.M.

Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett.

DOTTED-IN CURVES REPRESENT STOCK PUMP SELECTIONS

TOTAL HEAD IN FEET
CAPACITY IN U.S. GALLONS PER MINUTE
0
10
20
30
0
10
20
30
40
50
60
5¼
4 11/16
4"
3 7/16
30%
35%
40%
45%
48%
60-11
45%
40%
35%
⅓ HP
¼ HP

COLLECTOR LOOP

COMPRESSION TANK SELECTION WORKSHEET

Line 1. Determine Volume Of Fluid In Piping:

SIZE	:	LENGTH	x	GAL/FT	=	VOLUME	
(from System Layout Drawing)		(from System Layout Drawing)		(from Table 8a of DATA section)			
1¼"	:	(40+47+5+8) = 100	x	0.078	=	7.80	gal.
1"	:	(28+28+36+28) = 120	x	0.045	=	5.40	gal.
3/4"	:	8'	x	0.028	=	0.22	gal.
3/8"	:	(18x2) = 36'	x	0.007	=	0.25	gal.
	:		x		=		gal.
					TOTAL	13.67	gal.

Line 2. Determine Volume Of Fluid In The Following Components:

a. Collectors:
(Collector Fluid Capacity, from Engineering Handbook data 0.3) x
(Number Of Collectors, from Line 1d of Step 4A Worksheet 18) = 5.40 gal.

b. Heat Exchanger — shell (from Step 3A Worksheet):
(No. of Exchangers 2) x (Shell Length 29) x (Gal/Linear Inch .036) = 2.09 gal.

c. Purge Coil (from Step 3B Worksheet specifications) .... 1.03 gal.

d. Pump (from pump specifications) .... 0.50 gal.

e. Strainer (from strainer specifications) .... 0.50 gal.

f. Air Separator (from air separator specifications) .... 1.00 gal.

TOTAL 10.52 gal.

Line 3. Total Volume Of Fluid: (Line 1 TOTAL 13.67) + (Line 2 TOTAL 10.52) = 24.19 gal.

Line 4. Correct Total Volume for Ethylene Glycol:
(Line 3 24.19) x (Correction Factor from Table 3c of DATA section 1.5) = 36.28 gal.
NOTE: Maximum design temperature of most typical systems is 200°F. Enter Table 3c at 200°.

Line 5. Determine Tank Capacity Requirements Based on Line 4 figure
(from Table 8b of DATA section) .... 5.3 gal.
NOTE: Enter Table 8b at 200°F.

Line 6. Select Compression Tank (from Table 8c of DATA section):
Model no. 15, ASME CONSTRUCTION Gal. capacity 15

SIZING LOAD SIDE LOOPS

Load Side Loop Identification:

Heat Exchanger to Solar Heat Coil Loop = Segments Ⓐ, Ⓑ, Ⓒ, Ⓓ, Ⓔ, Ⓕ

Heat Exchanger to Storage Loop= Segments Ⓐ, Ⓑ, Ⓖ, Ⓗ, Ⓘ, Ⓕ

Storage to Solar Heat Coil Loop =Segments Ⓙ, Ⓒ, Ⓓ, Ⓔ, Ⓘ, Ⓗ, Ⓖ, Ⓚ

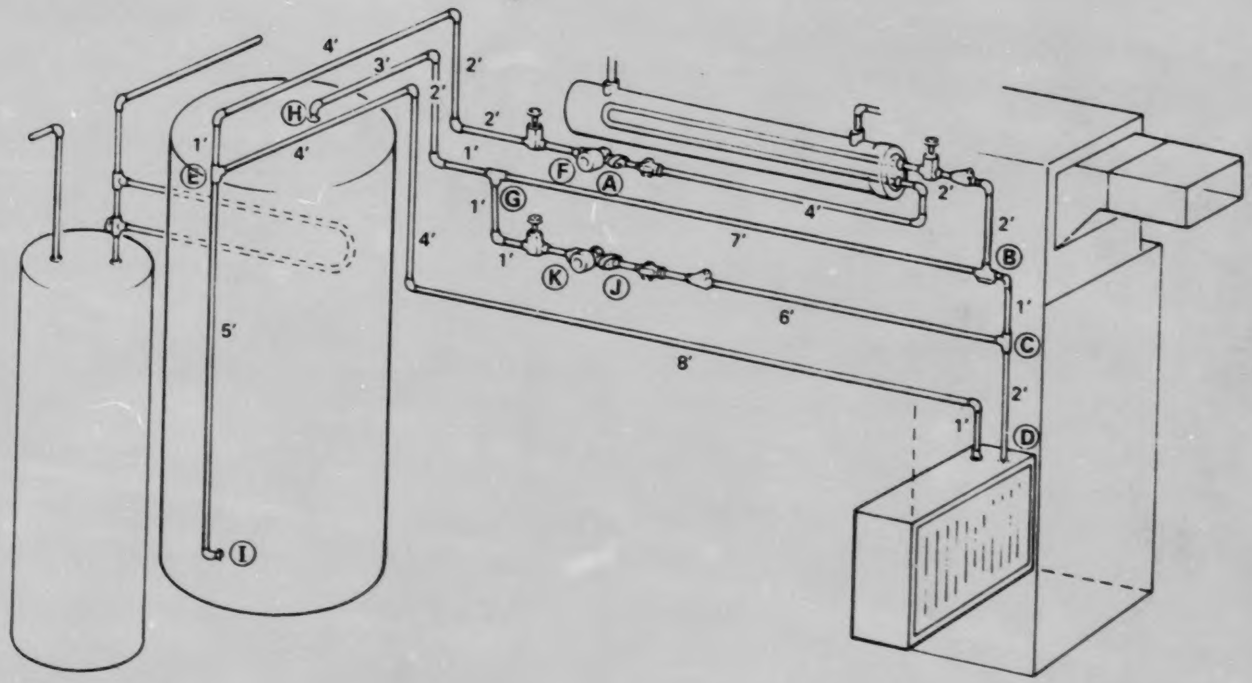
NOTE: Where there is bidirectional fluid flow in piping segments that contain vertical runs, size pipe according to a downrunner consideration.

PIPE LENGTHS

AB = 8′
BC = 1′
CD = 2′

DE = 17′
EF = 9′
BG = 7′

GH = 6′ (bidirectional & vertical, size to downrunner)
IE = 5′ (bidirectional & vertical, size to downrunner)
JC = 6′
GK = 2′



LOAD SIDE LOOPS

FLUID FLOW WORKSHEET

Line 1a. Total Effective Absorber Area Of Collector Array .... 277 sq. ft.

b. Solar Radiation Incidence (from Table 9 of DATA section) .... 220 Btuh/sq. ft.

c. Solar Load To Be Transferred Through Heat Exchanger:
(Line 1a 277) x (Line 1b 220) = ... 60940 Btuh

Line 2. Temperature Drop Across Heat Exchanger:

(Line 1c 60940)

(Line 3d of Collector Loop Fluid Flow Worksheet 8.4) x 444 (Heat carrying factor for glycol) = ... 16.34 °F

NOTE: Temperature drop across both shell and tube sides of heat exchanger is the same.

Line 3. Flow Rate For Load Side Loops:

(Line 1c 60940)

(Line 2 16.34) x 500 (heat carrying factor for water) = ... 7.46 OR 7.5 gpm

MASTER HYDRONICS WORKSHEET

FOR LOAD SIDE -- HEAT EXCH. TO SOLAR HEAT COIL LOOP
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION AB							
DOWNRUNNER PIPE 1	7.5	1	2.9	8	8.0	8.0	0.32
ELS (90° L.R) 3	"	"	"	—	1.7	5.1	.20
GATE VALVE 1	"	"	"	—	1.0	1.0	.04
CHECK VALVE (SWING) 1	"	"	"	—	10.0	10.0	.40
HEAT EXCHANGER 1		(FROM STEP 3A WORKSHEET) →					5.46
NOTE: BALANCE VALVE FRICTION LOSS AND PRESETTING FIGURED IN STEP 4C.							6.42
SECTION BC							
DOWNRUNNER PIPE 1	7.5	1	2.9	1	1.0	1.0	.04
EL (90° L.R.) 1	"	"	"	—	1.7	1.7	.07
3-WAY VALVE (DIVERT) 1		(CV=13, FROM TABLE 6C) →					.90
							1.11
SECTION CD							
DOWNRUNNER PIPE 1	7.5	1	2.9	2	2.0	2.0	.08
TEE (STR FLOW) 1	"	"	"	—	1.7	1.7	.07
							.15
SECTION DE							
PIPE 1	7.5	1¼	1.8	17	17.0	17.0	.27
ELS (90° L.R.) 3	"		"	—	2.3	6.9	.10
SOLAR COIL 1		(FROM STEP 3D WORKSHEET) →					6.00
							6.37
						TOTAL	14.05

MASTER HYDRONICS WORKSHEET

FOR LOAD SIDE -- HEAT EXCH. TO SOLAR COIL LOOP (CONTINUED)

USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION EF							
DOWNRUNNER PIPE 1	7.5	1	2.9	9	9.0	9.0	0.36
SUDDEN CONTRACTION (3/4) 1	"	1¼→1	"	—	.7	.7	.03
TEE (BRANCH FLOW) 1	"	1	"	—	5.0	5.0	.20
ELS (90° L.R.) 3	"	"	"	—	1.7	5.1	.20
GATE VALVE 1	"	"	"	—	1.0	1.0	.04
							.83
						TOTAL	.83

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MASTER HYDRONICS WORKSHEET

FOR LOAD SIDE -- HEAT EXCH. TO STORAGE LOOP
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION AB		(FIGURED PREVIOUSLY) →					6.42
SECTION BG							
PIPE 1	7.5	1¼	1.8	7	7.0	7.0	.11
SUDDEN ENLARGEMENT (3/4) 1	"	1→1¼	"	—	.7	.7	.03
3-WAY VALVE (DIVERT) 1		(CV=13, FROM TABLE 6C) →					.90
							1.04
SECTION GH							
BIDIRECTIONAL DOWNRUNNER PIPE 1	7.5	1	2.9	6	6.0	6.0	.24
SUDDEN CONTR. (3/4) 1	"	1¼→1	"	—	.7	.7	.03
TEE (STRAIGHT FLOW) 1	"	1	"	—	1.7	1.7	.07
ELS (90° L.R.) 3	"	"	"	—	1.7	5.1	.20
							.54
SECTION H							
STORAGE TANK ENTRANCE LOSS (SHARP EDGE) 1	7.5	1	2.9	—	3.7	3.7	.15
STORAGE TANK EXIT LOSS (SHARP) 1	"	"	"	—	1.8	1.8	.07
							.22
SECTION IE							
BIDIRECTIONAL DOWNRUNNER PIPE 1	7.5	1	2.9	5	5.0	5.0	.20
EL (90° L.R.) 1	"	"	"	—	1.7	1.7	.07
							.27
SECTION EF		(FIGURED PREVIOUSLY) →					.83
						TOTAL	9.32

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MASTER HYDRONICS WORKSHEET

FOR LOAD SIDE -- STORAGE TO SOLAR COIL LOOP
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)	Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION JC							
1 PIPE	7.5	1¼	1.8	6	6.0	6.0	0.10
1 CHECK VALVE (SWING)	"	"	"	—	14.0	14.0	.22
NOTE: BALANCE VALVE FRICTION LOSS & PRESETTING FIGURED IN STEP 4C.							.32
SECTION CD							
1 DOWN RUNNER PIPE	7.5	1	2.9	2	2.0	2.0	.08
1 SUDDEN CONTR. (3/4)	"	1¼→1	"	—	.7	.7	.03
1 TEE (BRANCH FLOW)	"	1	"	—	5.0	5.0	.20
							.31
SECTION DE		(FIGURED PREVIOUSLY) →					6.37
SECTION EI		(FIGURED PREVIOUSLY) →					.27
SECTION IH		(FIGURED PREVIOUSLY) →					.22
SECTION HG		(FIGURED PREVIOUSLY) →					.54
		(CONTINUED)					
						TOTAL	8.03

MASTER HYDRONICS WORKSHEET

FOR LOAD SIDE -- STORAGE TO SOLAR COIL LOOP (CONTINUED)
USED TO SIZE PIPE & DETERMINE FRICTION LOSS

Col. A SECTION & COMPONENT (from System Layout Drawing)		Col. B FLUID FLOW (gpm) (from Line 3 of Step 4A Worksheet)	Col. C PIPE SIZE (in., I.D.) (from Table 4 of DATA section) NOTE—Enter size on System Layout Drawing.	Col. D VELOCITY (fps) (from Table 4 of DATA section)	Col. E PIPE LENGTH (ft.) (from System Layout Drawing)	Col. F EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.) (from Tables 5a, 5b or 5c of DATA Section)	Col. G TOTAL EQUIVALENT LENGTH OF STANDARD FITTINGS (ft.): (Col. A) x (Col. F)	Col. H. FRICTION LOSS (ft of head) (from Tables 6a, 6b or 6c of DATA section or Step 3A, 3B, 3C or 3D Worksheets
SECTION GK								
DOWNRUNNER PIPE	1	7.5	1	2.9	2	2.0	2.0	.08
TEE (BRANCH FLOW)	1	"	"	"	—	5.0	5.0	.20
EL (90° L.R.)	1	"	"	"	—	1.7	1.7	.07
GATE VALVE	1	"	"	"	—	1.0	1.0	.04
								.39

TOTAL .39

LOAD SIDE LOOPS
FLOW BALANCE WORKSHEET

Line 1. Friction Losses (from Step 4B Worksheets):

a. Heat Exchanger to Solar Heat Coil Loop .... 14.88 ft. of head

b. Heat Exchanger to Storage Loop .... 9.32 ft. of head

c. Storage To Solar Heat Coil Loop .... 8.42 ft. of head

NOTE: Since one pump operates both heat exchanger to solar coil and heat exchanger to storage loops, each is actually a branch flow path. As in the collector loop sizing procedure, the critical path must be determined to properly select the pump. Unlike the collector loop, the non-critical path flow is not balanced. This is because the increased flow rate will result in a heat transfer advantage that outweighs installing a balance valve.

The storage to solar coil loop has its own pump and only one flow path, which is considered the critical path.

Critical Paths: Heat Exch. to Coil Loop
Storage to Solar Coil Loop

Line 2. Determine Presettings Of Critical Path Balance Valves:

a. (Line 1 Critical Path Friction Loss 14.88) x 30% = ... 4.46 ft. of head

b. Use Circuit Setter Calculator—Preset Side to determine setting needed to produce Line 2a friction loss: .... 1" B.V., 5 ° closed

c. (Line 1 Critical Path Friction Loss 8.42) x 40% = ... 3.37 ft. of head

d. Use Circuit Setter Calculator—Preset Side to determine setting needed to produce Line 2c friction loss: .... 1¼" B.V., 26° ° closed

Line 3. Design Friction Losses:

a. (Line 1 Critical Path Friction Loss 14.88) + (Line 2a 4.46) = 19.34 ft. of head

b. (Line 1 Critical Path Friction Loss 8.42) + (Line 2c 3.37) = 11.79 ft. of head

LOAD SIDE LOOPS

PUMP SELECTION WORKSHEET

Heat Exchanger to Solar Coil & Heat Exchanger to Storage Loop Pump

Line 1a. Design Friction Loss (from Line 3a of Step 4C Worksheet) .... 19.34 ft. of head

b. Design Flow Rate (from Line 3 of Step 4A Worksheet) .... 7.5 gpm

NOTE: The intersection of the above numbers on a graph is the system design point.

Line 2. Select Pump Performance Curve graph (See Tables 7a or 7b in DATA section) whose range is such that the system design point falls near mid-range on graph.

Line 3. Calculate at least 4 points of System Performance Curve from the following formula:

$$\left(\frac{Q2}{Q1}\right)^2 = \frac{h2}{h1}$$

Where:

Q2 = final flow (arbitrary number)
Q1 = known flow (from Line 1b)
h2 = final friction loss (this is to be determined)
h1 = known friction loss (from Line 1a)

$\left(\frac{3}{7.5}\right)^2 = \frac{h2}{19.34}$ h2 = 3.09

$\left(\frac{5}{7.5}\right)^2 = \frac{h2}{19.34}$ h2 = 8.6

$\left(\frac{10}{7.5}\right)^2 = \frac{h2}{19.34}$ h2 = 34.38

$\left(\frac{12}{7.5}\right)^2 = \frac{h2}{19.34}$ h2 = 49.51

Line 4. Plot the system design point and the 4 (Q2, h2) System Curve points calculated in Line 3 on Pump Performance Curve graph. (See example on next page.)

Line 5. Select pump whose performance curve is nearest the system design point. If the design point is equidistant between two pump curves, normally size to the higher pump curve. The system design point should also intersect the pump curve in the middle one-third range of pump curve, or slightly left of middle.

Pump Selected: rpm 1750 hp 1/4 impeller 4 11/16 model no. 1AA, SERIES 60

Storage to Solar Coil Loop Pump

Line 6a. Design Friction Loss (from Line 3b of Step 4C Worksheet) .... 11.79 ft. of head

b. Design Flow Rate (from Line 3 of Step 4A Worksheet) .... 7.5 gpm

NOTE: The intersection of the above numbers on a graph is the system design point.

Line 7. Select Pump Performance Curve graph (see Tables 7a or 7b in DATA section) whose range is such that the system design point falls near mid-range on graph.

Line 8. Calculate at least 4 points of System Performance Curve (use formula from Line 3 with one exception; h1 is from Line 6a instead of Line 1a):

$\left(\frac{3}{7.5}\right)^2 = \frac{h2}{11.79}$ h2 = 1.89

$\left(\frac{5}{7.5}\right)^2 = \frac{h2}{11.79}$ h2 = 5.24

$\left(\frac{8}{7.5}\right)^2 = \frac{h2}{11.79}$ h2 = 13.41

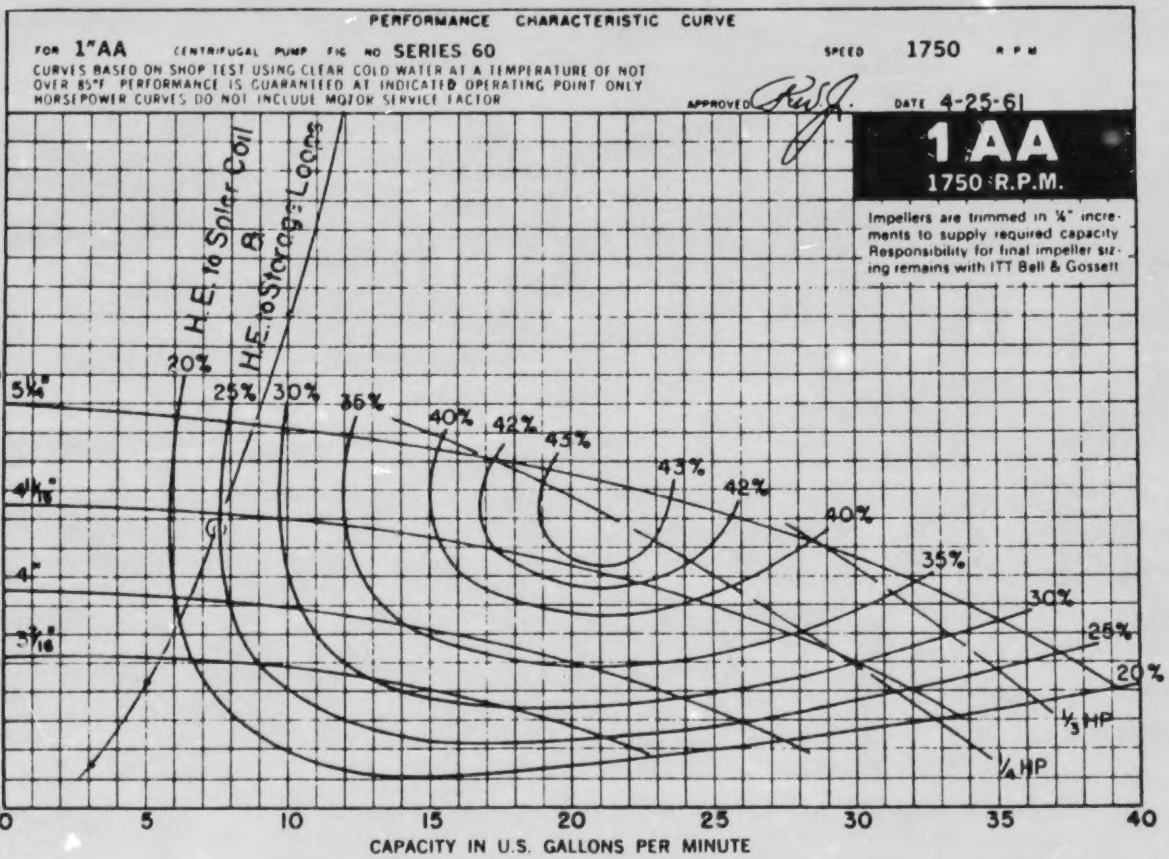
$\left(\frac{10}{7.5}\right)^2 = \frac{h2}{11.79}$ h2 = 20.96

Line 9. Plot the system design point and the 4 (Q2, h2) System Curve points calculated in Line 8 on Pump Performance Curve graph. (See example on next page.)

Line 10. Select pump using same rationale as in Line 5 above.

Pump Selected: rpm 3200 hp — impeller — model no. UP 26-64

PUMP PERFORMANCE CURVES — ITT BELL & GOSSETT



PERFORMANCE CHARACTERISTIC CURVE

FOR 1"AA CENTRIFUGAL PUMP FIG NO SERIES 60 SPEED 1750 RPM

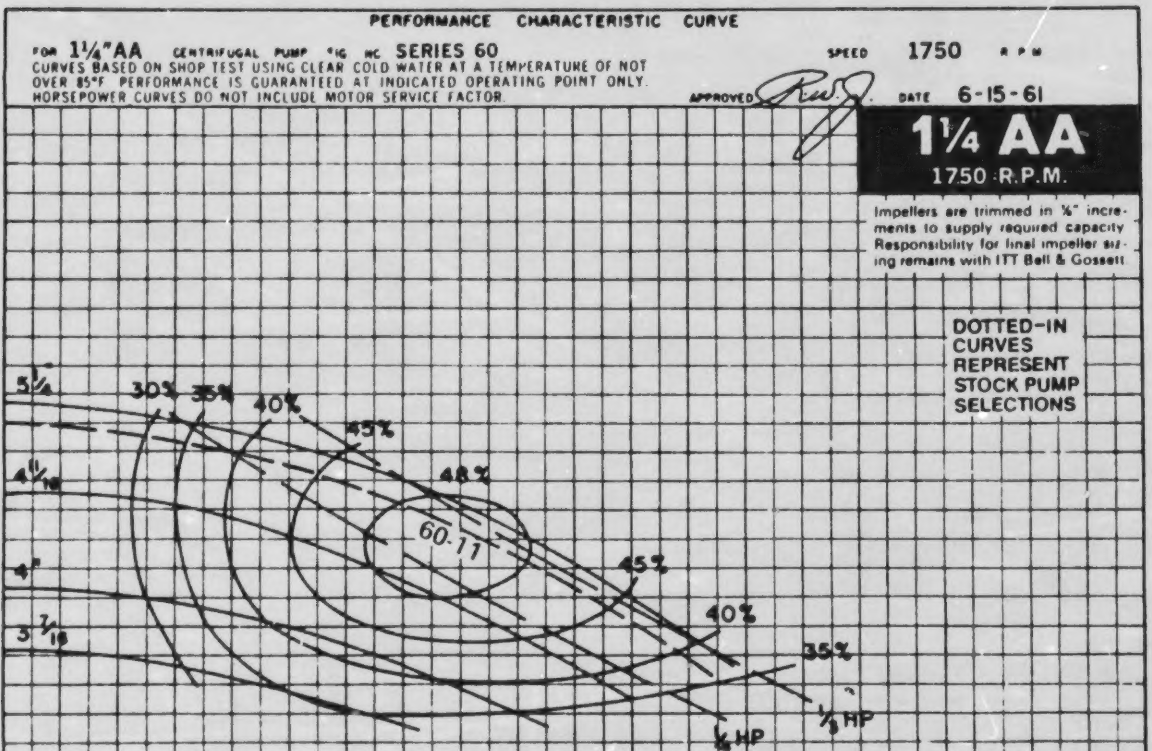
CURVES BASED ON SHOP TEST USING CLEAR COLD WATER AT A TEMPERATURE OF NOT OVER 85°F PERFORMANCE IS GUARANTEED AT INDICATED OPERATING POINT ONLY HORSEPOWER CURVES DO NOT INCLUDE MOTOR SERVICE FACTOR

APPROVED DATE 4-25-61

1 AA

1750 R.P.M.

Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett



PERFORMANCE CHARACTERISTIC CURVE

FOR 1¼"AA CENTRIFUGAL PUMP FIG NO SERIES 60 SPEED 1750 RPM

CURVES BASED ON SHOP TEST USING CLEAR COLD WATER AT A TEMPERATURE OF NOT OVER 85°F. PERFORMANCE IS GUARANTEED AT INDICATED OPERATING POINT ONLY. HORSEPOWER CURVES DO NOT INCLUDE MOTOR SERVICE FACTOR.

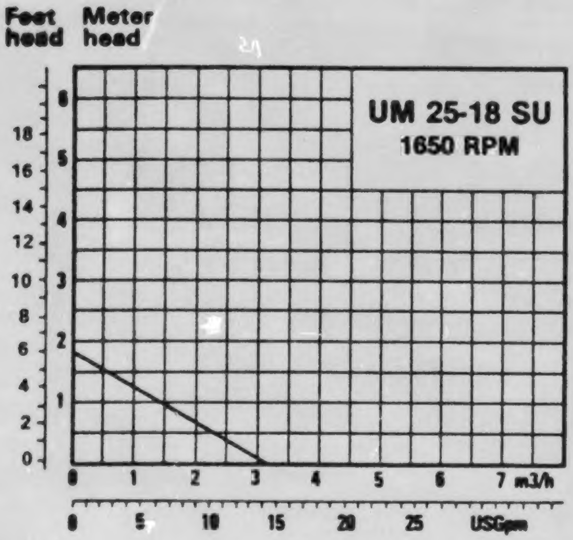
APPROVED DATE 6-15-61

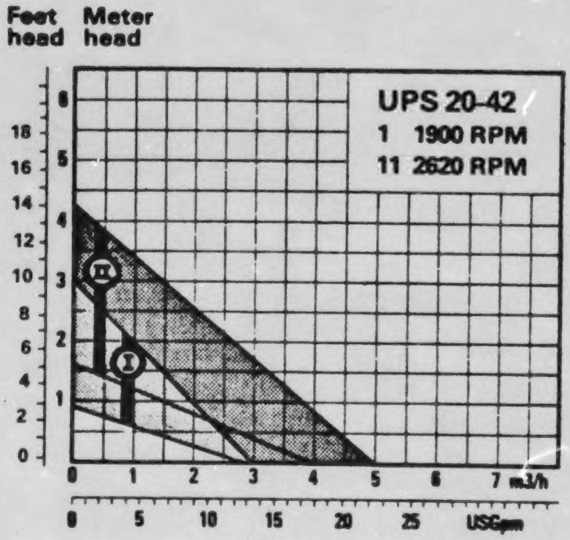
1¼ AA

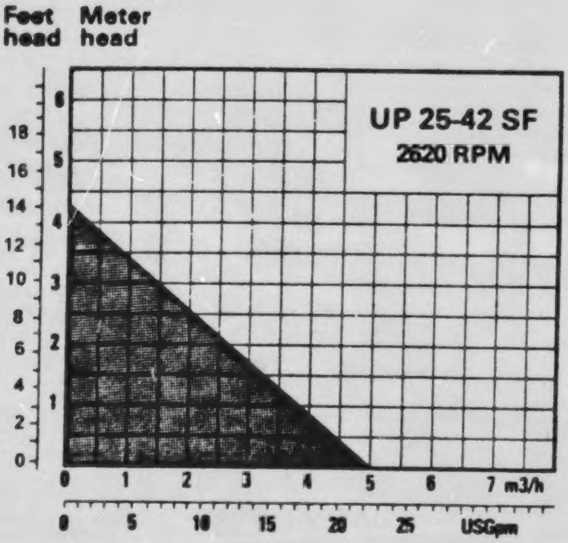
1750 R.P.M.

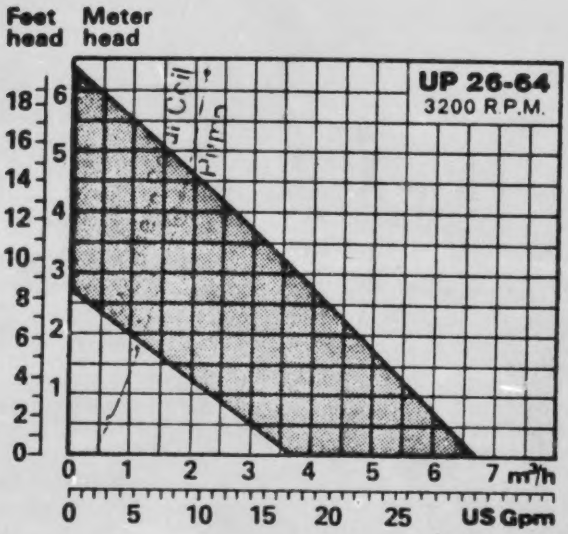
Impellers are trimmed in ⅛" increments to supply required capacity. Responsibility for final impeller sizing remains with ITT Bell & Gossett

DOTTED-IN CURVES REPRESENT STOCK PUMP SELECTIONS


Feet head
Meter head
UM 25-18 SU
1650 RPM
0
2
4
6
8
10
12
14
16
18
1
2
3
4
5
6
0
1
2
3
4
5
6
7 m3/h
0
5
10
15
20
25
USGpm


Feet head
Meter head
UPS 20-42
1 1900 RPM
11 2620 RPM
II
I
0
2
4
6
8
10
12
14
16
18
1
2
3
4
5
6
0
1
2
3
4
5
6
7 m3/h
0
5
10
15
20
25
USGpm

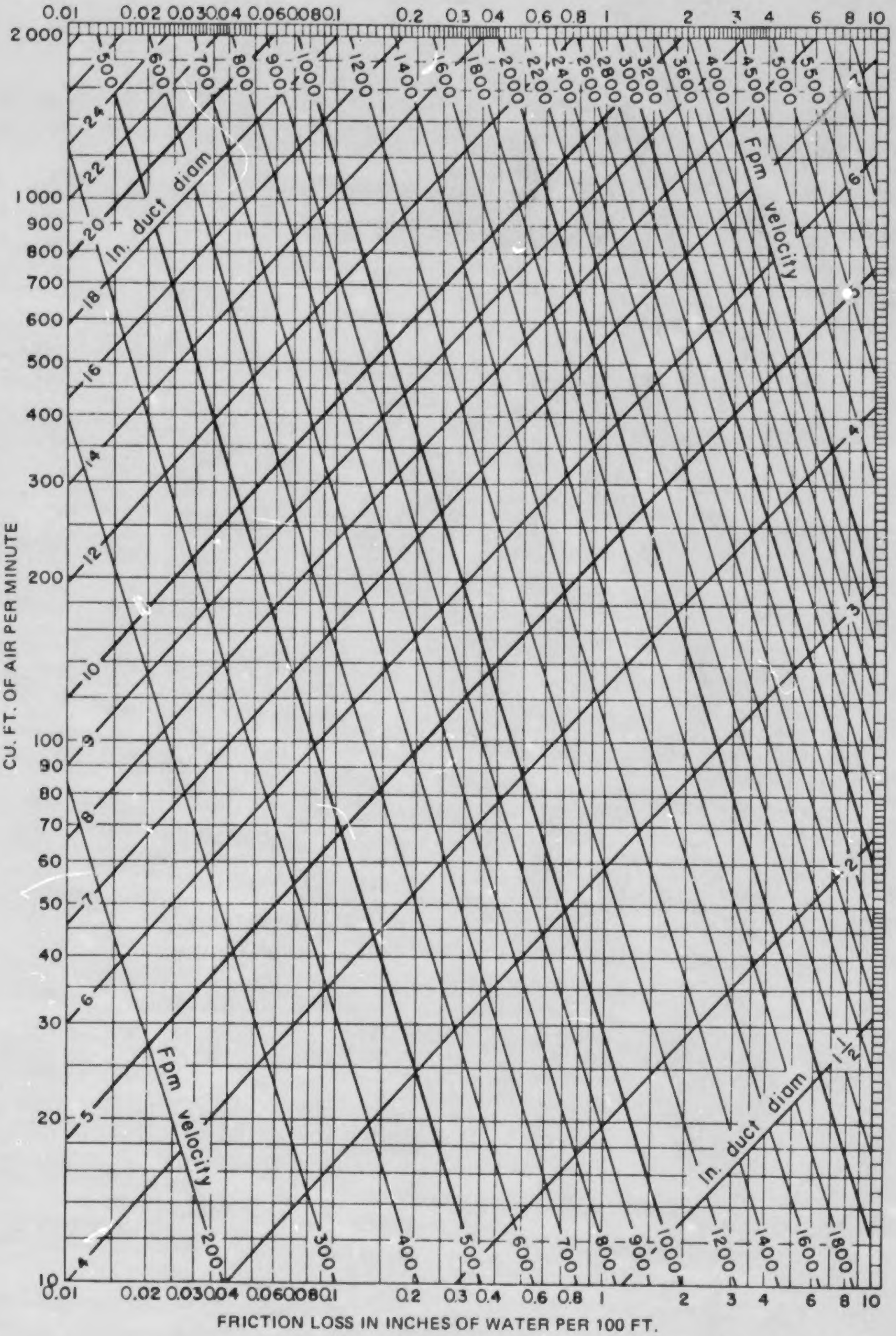

Feet head
Meter head
UP 25-42 SF
2620 RPM
0
2
4
6
8
10
12
14
16
18
1
2
3
4
5
6
0
1
2
3
4
5
6
7 m3/h
0
5
10
15
20
25
USGpm


Feet head
Meter head
UP 26-64
3200 R.P.M.
0
2
4
6
8
10
12
14
16
18
1
2
3
4
5
6
0
1
2
3
4
5
6
7 m3/h
0
5
10
15
20
25
US Gpm

As for air systems, the sizing process occurs much in the same manner as the liquid system. A friction loss chart can be used to size the duct, when a manufacturer's worksheet is not available. Using 2 CFM per ft^2 of collector array, the collector loop air requirement can be established and the duct size selected from the friction loss chart. Don't forget to add the equivalent length of the fittings (tees, elbows, etc.). Then the air handler can be selected from performance charts or curves.

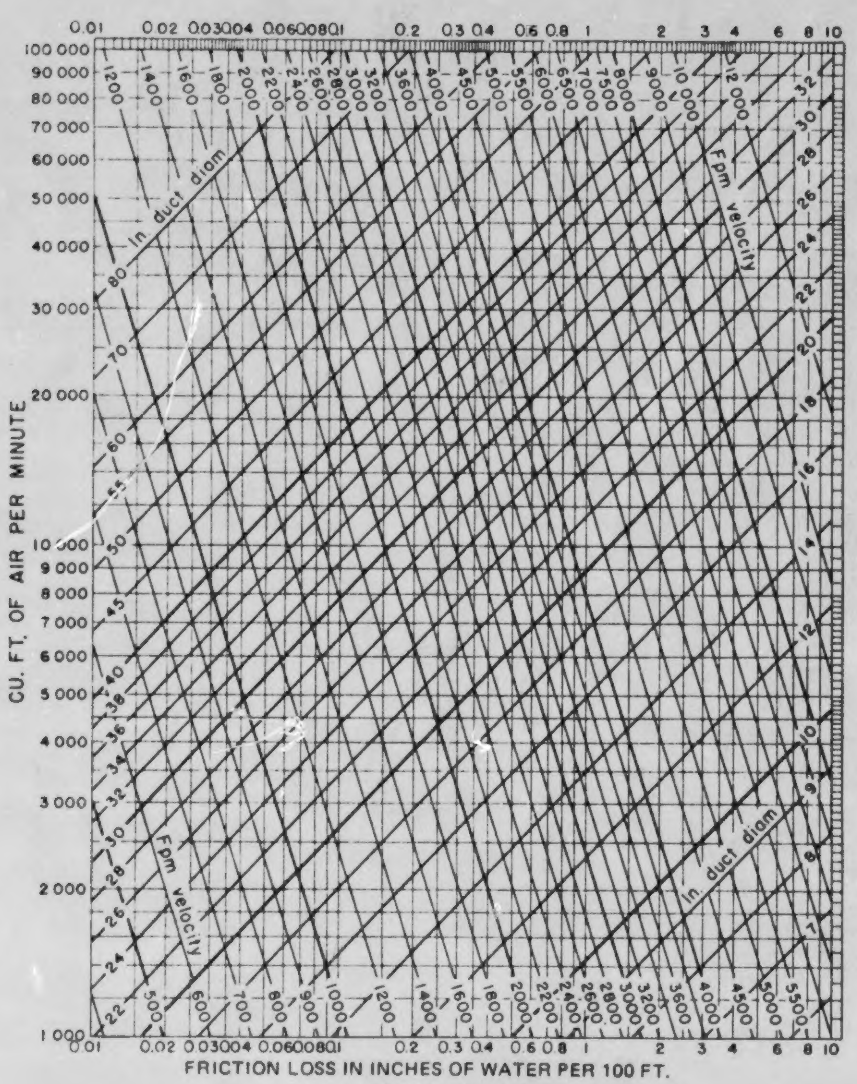
The load side duct can be sized in the same manner. If the air requirements for the space are considerably different from those of the collector loop, then additional blowers and dampers must be used, thus increasing the system cost.

The manufacturer's worksheet is a reliable method for sizing the air requirements.



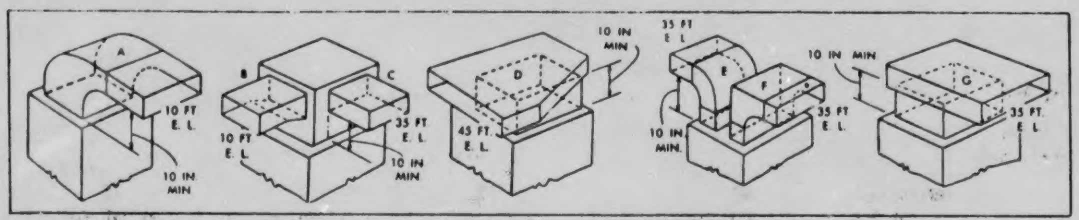
Friction loss chart

(From ASHRAE Guide, 1965, by permission)

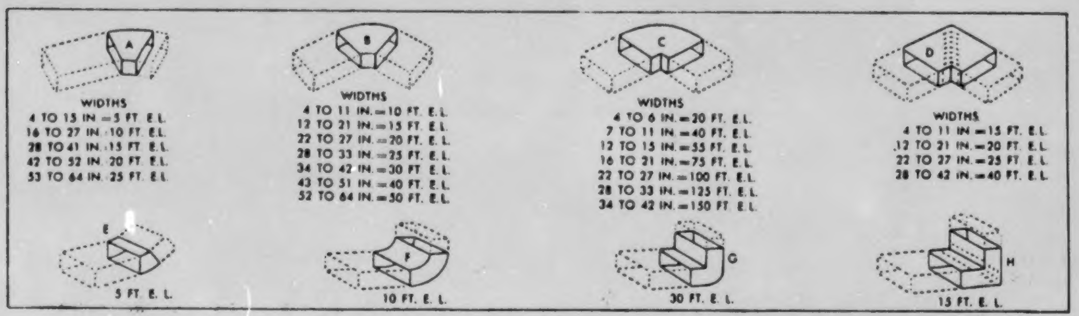


Friction loss chart

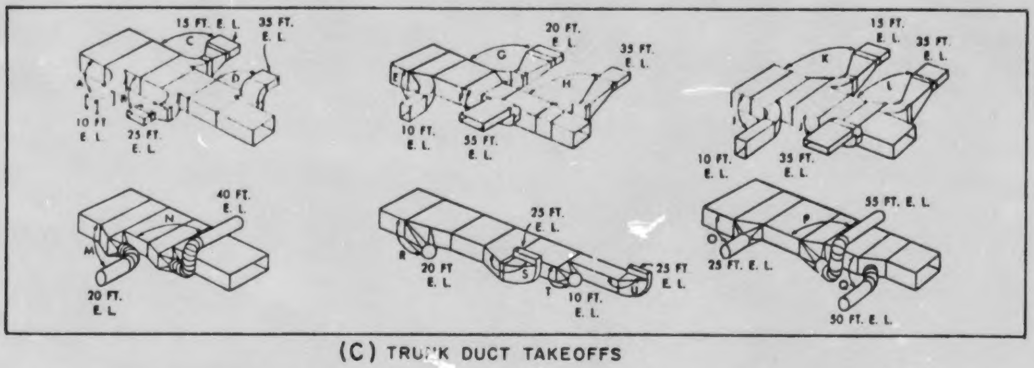
(From ASHRAE Guide, 1965, by permission)


A
10 FT E. L.
10 IN MIN
B
C
10 FT E. L.
35 FT E. L.
10 IN MIN
10 IN MIN
D
45 FT. E. L.
35 FT E. L.
E
F
10 IN. MIN.
35 FT E. L.
10 IN MIN
G
35 FT. E. L.

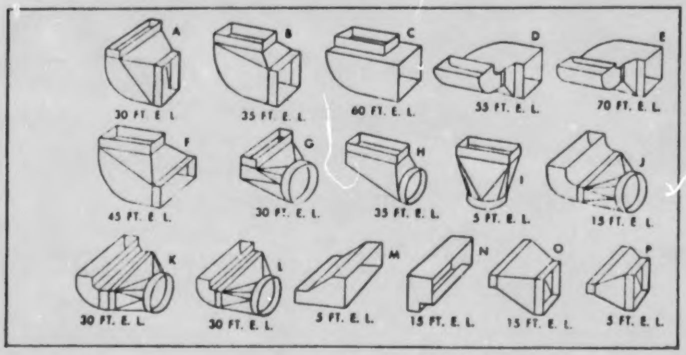
(A) WARM AIR AND RETURN AIR BONNET OR PLENUM
EQUIVALENT LENGTHS BASED ON 8" DEPTH OF DUCT


A
WIDTHS
4 TO 15 IN.=5 FT. E.L.
16 TO 27 IN.=10 FT. E.L.
28 TO 41 IN.=15 FT. E.L.
42 TO 52 IN.=20 FT. E.L.
53 TO 64 IN.=25 FT. E.L.
B
WIDTHS
4 TO 11 IN.=10 FT. E.L.
12 TO 21 IN.=15 FT. E.L.
22 TO 27 IN.=20 FT. E.L.
28 TO 33 IN.=25 FT. E.L.
34 TO 42 IN.=30 FT. E.L.
43 TO 51 IN.=40 FT. E.L.
52 TO 64 IN.=50 FT. E.L.
C
WIDTHS
4 TO 6 IN.=20 FT. E.L.
7 TO 11 IN.=40 FT. E.L.
12 TO 15 IN.=55 FT. E.L.
16 TO 21 IN.=75 FT. E.L.
22 TO 27 IN.=100 FT. E.L.
28 TO 33 IN.=125 FT. E.L.
34 TO 42 IN.=150 FT. E.L.
D
WIDTHS
4 TO 11 IN.=15 FT. E.L.
12 TO 21 IN.=20 FT. E.L.
22 TO 27 IN.=25 FT. E.L.
28 TO 42 IN.=40 FT. E.L.
E
5 FT. E. L.
F
10 FT. E. L.
G
30 FT. E. L.
H
15 FT. E. L.

(B) ANGLES AND ELBOWS FOR TRUNK DUCTS
EQUIVALENT LENGTHS BASED ON 8" DEPTH OF DUCT


15 FT. E. L.
35 FT. E. L.
C
D
A
B
10 FT E. L.
25 FT. E. L.
20 FT. E. L.
35 FT. E. L.
G
E
F
H
I
10 FT. E. L.
55 FT. E. L.
15 FT. E. L.
35 FT. E. L.
K
J
L
10 FT. E. L.
35 FT. E. L.
40 FT. E. L.
N
M
20 FT. E. L.
25 FT. E. L.
R
20 FT E. L.
S
T
10 FT. E. L.
U
25 FT E. L.
55 FT. E. L.
P
O
25 FT. E. L.
Q
50 FT. E. L.

(C) TRUNK DUCT TAKEOFFS


A
30 FT. E. L.
B
35 FT. E. L.
C
60 FT. E. L.
D
55 FT. E. L.
E
70 FT. E. L.
F
45 FT. E. L.
G
30 FT. E. L.
H
35 FT. E. L.
I
5 FT. E. L.
J
15 FT. E. L.
K
30 FT. E. L.
L
30 FT. E. L.
M
5 FT. E. L.
N
15 FT. E. L.
O
15 FT. E. L.
P
5 FT. E. L.

(D) BOOT FITTINGS
FROM BRANCH TO STACK

Equivalent lengths for various types of fittings

V-S-230

SPECIFICATIONS

Model Number		G12Q3-82	G12Q3-110	*G12Q4-110	G12Q3-137	G12Q4-137	G12Q5-137	G12Q5-165
Btuh input		82,000	110,000	110,000	137,000	137,000	137,000	165,000
Btuh bonnet output		65,600	88,000	88,000	109,600	109,600	109,600	132,000
Flue size (in.)		4	5	5	5 (6 opt.)	5 (6 opt.)	5 (6 opt.)	6 oval
High static certified by A.G.A. (in. wg.)		.50	.50	.50	.50	.50	.50	.50
Gas piping size (I.P.S. in.)	Natural	1/2	1/2	1/2	1/2	1/2	1/2	3/4
	†Propane	1/2	1/2	1/2	1/2	1/2	1/2	3/4
Blower wheel nom. diam. x width (in.)		10 x 7	10 x 8	10 x 8	10 x 8	12 x 9	12 x 12	12 x 12
Blower motor horsepower		1/3	1/3	1/2	1/3	1/3	3/4	3/4
Net filter area (sq. ft.) & cut size (in.)		(5.8) 36 x 28 x 1	(6.6) 40 x 26 x 1	(6.6) 40 x 28 x 1	(8.9) 52 x 28 x 1	(8.9) 52 x 28 x 1	(8.9) 52 x 28 x 1	(9.2) 54 x 28 x 1
Tons of cooling that can be added		2, 2-1/2 or 3	2, 2-1/2 or 3	3-1/2 or 4	2-1/2 or 3	3-1/2 or 4	3-1/2, 4 or 5	3-1/2, 4 or 5
Net weight (lbs.)		166	192	200	236	252	262	301
Number of packages in shipment		1	1	1	1	1	1	1
Electrical characteristics		120 volt – 60 hertz – 1 phase (All Units)						
Return Air Cabinet	Model No.	RA10-16-49	RA-10-16-49	RA10-16-49	RA10-16-53	RA10-16-53	RA10-16-53	RA10-16-53
	Net Weight (lbs.)	65	65	65	75	75	75	75

NOTE – High altitude derate – A.G.A. requires gas furnaces be derated 4% per thousand feet above sea level when the installation is at an altitude of 2000 feet or more. Thus an installation at 3000 feet altitude requires a derate of 12% while up to 2000 feet altitude the unit has a full rating.

*A.G.A. Certified as a 45° to 75° rise furnace only. †Changeover Kit is furnished for field conversion to propane.

Model Number		*G12D2-55	G12D2-82	G12D-110
Btuh input		55,000	82,000	110,000
Btuh bonnet output		44,000	65,600	88,000
Flue size (in.)		4	4	5
High static certified by A.G.A. (in. wg.)		.50	.50	.20
Gas piping size (I.P.S. In.)	Natural	1/2	1/2	1/2
	†Propane	1/2	1/2	1/2
Blower wheel nominal diam. x width (in.)		9 x 7	9 x 7	9 x 9
Blower motor horsepower		1/5	1/5	1/8
Number and size of filters (in.)		1–16 x 25 x 1	1–16 x 25 x 1	1–16 x 25 x 1
Tons of cooling that can be added		1-1/2 or 2	1-1/2 or 2	----
Net weight (lbs.)		143	158	185
Number of packages in shipment		1	1	1
Electrical characteristics		120 volt–60 hertz–1 phase (All Units)		
Return Air Cabinet	Model No.	RA10-16-49	RA10-16-49	RA10-16-49
	Net Weight (lbs.)	65	65	65

NOTE – High altitude derate – A.G.A. requires gas furnaces be derated 4% per thousand feet above sea level when the installation is at an altitutde of 2000 feet or more. Thus an installation at 3000 feet altitude requires a derate of 12% while up to 2000 feet altitude the unit has a full rating.

*A.G.A. Certified as a 45° to 75° rise furnace only.

†Changeover Kit is furnished for field conversion to propane.

A.G.A. INSTALLATION CLEARANCES

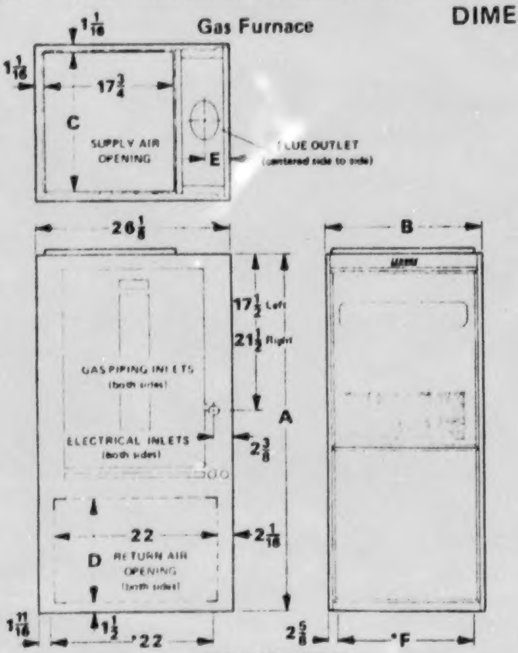
Sides	1 inch
Rear	1 inch
Top	1 inch
Front	6 inches
Floor	Combustible
*Flue	*6 inches

*This is clearance to all flue pipes except type "B". Type "B" flue clearance is as listed by U.L.

When a furnace is installed in a confined space, two openings must be provided into the confined area, one opening near the top of the enclosure and one near the bottom. Each opening shall have at least one square inch of free area per 1000 Btuh of input and must not be smaller than 100 square inches of free area.

DIMENSIONS (inches)

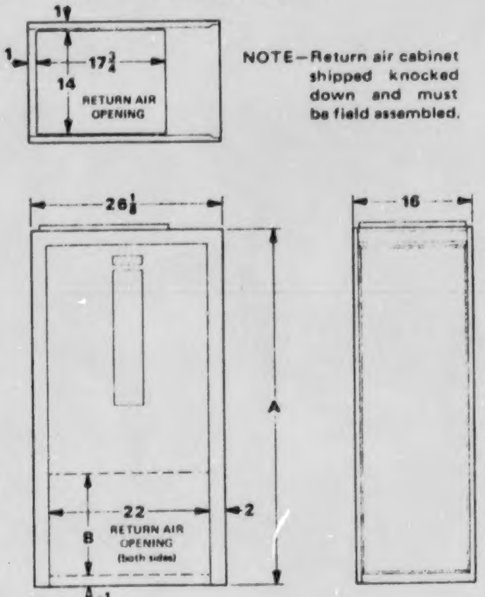
Gas Furnace


1 1/16
1 1/16
17 3/4
C
SUPPLY AIR OPENING
FLUE OUTLET (centered side to side)
E
26 1/8
B
17 1/2 Left
21 1/2 Right
GAS PIPING INLETS (both sides)
A
ELECTRICAL INLETS (both sides)
2 3/8
22
2 1/16
D
RETURN AIR OPENING (both sides)
1 11/16
1 1/2
*22
2 5/8
*F

*Bottom return air knockout

Model No.	A	B	C	D	E	F
G12D2 55 G12D2-82, G12Q3-82	49	16 1/4	14-1/8	14	3-1/8	11
G12D-110 G12Q3-110, G12Q4-110	49	21-1/4	19-1/8	14	3-1/8	16
G12Q3-137 G12Q4-137, G12Q5-137	53	26-1/4	24-1/8	18	3-3/8	21
G12Q5-165	53	31-1/4	29-1/8	18	3-3/8	26

Return Air Cabinet



NOTE—Return air cabinet shipped knocked down and must be field assembled.

Model No.	RA10-16-49	RA10-16-53
A	49	53
B	14	18

235

9c

G12D2-55 BLOWER PERFORMANCE

External Static Pressure (in. wq.)	Air Volume (cfm) @ Various Speeds		
	High	Medium	Low
0	990	750	575
05	970	745	575
10	945	735	575
15	925	725	570
20	900	710	565
25	875	695	555
.30	850	680	545
.40	795	635	510
.50	730	570	455

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q3-82 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds			
	High	Med-High	Med-Low	Low
0	1535	1280	1000	825
.05	1510	1265	995	825
.10	1485	1255	990	825
.15	1460	1240	980	820
.20	1435	1225	970	820
.25	1410	1210	965	815
30	1385	1190	955	805
40	1330	1155	930	780
50	1270	1110	890	750
60	1205	1050	840	715
.70	1130	980	775	665
.80	1040	890	695	600

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q4-110 BLOWER PERFORMANCE

External Static Pressure (in. wg.)	Air Volume (cfm) @ Various Speeds		
	High	Medium	Low
0	1875	1475	1180
.05	1845	1465	1180
.10	1815	1450	1180
.15	1785	1440	1180
.20	1755	1430	1180
25	1725	1415	1180
.30	1690	1400	1180
.40	1625	1370	1160
.50	1555	1330	1130
60	1475	1275	1085

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q3-137 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds		
	High	Medium	Low
0	1740	1410	1130
.05	1705	1400	1125
.10	1670	1380	1120
15	1630	1360	1115
20	1590	1340	1110
25	1550	1320	1100
.30	1510	1295	1085
.40	1435	1250	1060
.50	1355	1190	----

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q5-137 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds				
	High	Med-High	Medium	Med-Low	Low
0	2630	2300	2170	1805	1590
05	2590	2275	2135	1775	1570
10	2550	2245	2095	1750	1540
15	2510	2210	2060	1720	1520
.20	2465	2175	2030	1695	1490
.25	2425	2140	1995	1670	1460
.30	2385	2110	1960	1640	1430
.40	2300	2030	1885	1575	1370
.50	2260	1950	1820	1500	1300
.60	2130	1870	1745	1420	1240
.70	2040	1780	1665	1340	1175
80	1955	1690	1605	1240	1105

NOTE—All cfm is measured external to the unit with the air filter in place.

V-S-231

G12D2-82 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds		
	High	Medium	Low
0	1010	790	620
.05	985	780	620
.10	960	770	615
.15	930	760	610
.20	905	745	605
.25	875	730	595
.30	850	710	580
.40	790	660	---
.50	725	595	----

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q3-110 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds			
	High	Med-High	Med-Low	Low
0	1590	1330	1050	855
.05	1575	1325	1050	855
.10	1555	1315	1045	860
.15	1535	1305	1040	860
.20	1510	1290	1035	855
.25	1490	1280	1025	850
.30	1460	1260	1015	840
.40	1405	1225	990	820
.50	1340	1180	955	....
.60	1270	1125	910	....
.70	1190	1060	850	....
.80	1100	985	....	----

NOTE—All cfm is measured external to the unit with the air filter in place.

G12D-110 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm)
0	890
.05	880
.10	865
.15	850
.20	830

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q4-137 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds		
	High	Medium	Low
0	2200	1790	1510
.05	2155	1786	1520
.10	2110	1778	1525
.15	2066	1764	1523
.20	2020	1743	1520
.25	1970	1720	1510
.30	1922	1690	1493
.40	1820	1614	1450
.50	1702	1520	1382
.60	1570	1412	1296
.70	1403	1291	1150
.80	1180	1050	----

NOTE—All cfm is measured external to the unit with the air filter in place.

G12Q5-165 BLOWER PERFORMANCE

External Static Pressure (in. wg)	Air Volume (cfm) @ Various Speeds				
	High	Med-High	Medium	Med-Low	Low
0	2710	2450	2255	1860	1670
.05	2685	2425	2235	1845	1655
.10	2655	2395	2210	1825	1635
.15	2625	2370	2190	1810	1615
.20	2600	2340	2165	1790	1590
.25	2565	2315	2140	1770	1570
.30	2535	2285	2115	1745	1550
.40	2470	2230	2060	1695	1490
.50	2400	2170	2000	1640	1435
.60	2330	2100	1930	1580	1370
.70	2250	2025	1860	1515	1310
.80	2165	1930	1780	1440	1240
.90	2065	1825	1695	1360	....
1.00	1925	1710	1605	1270	....

NOTE—All cfm is measured external to the unit with the air filter in place.

SOLARON SYSTEM SIZING WORKSHEET – SOLAR WITH SPACE HEATING

Reference

Project Name SAMPLE CALCULATION Location DENVER, COLORADO Date OCT. 1978
Building Design Heat Loss 60,000 Btu/hr. At Design Temp Difference – ΔT = 69 °F
800.3 Building Heated Floor Area 3,000 Ft.^{2} (Avg. Heat Loss Per Ft.^{2} = 20 Btu/hr/ft.^{2}
800.4, 800.5 Latitude 39.8 ° Collector Tilt 53 ° Collector Orientation from Due South 20 ° (West) ~~(East)~~
800.2, 800.12-16 Desired Annual Fuel Savings 50 % ⇒ Solaron Conversion Factor = S.C.F. = $\boxed{52.4}$

Section 800 — Step 1 — Collector Area

800.7, 800.11 [60,000 $\frac{\text{BTU}}{\text{HR}}$) ÷ (69 °F ΔT)] ÷ [52.4] (S.C.F.) = (16.6) (no. of panels) x (18.72) = 310.8 Ft.^{2} ÷ (1.0) (tilt corrections) ÷ (0.97) (orientation corrections) = 320.4 ft.^{2} (calculated area)

200.8, 200.9 Collector Array 2 Panels High 9 Panels Wide (Vertical) ~~(Horizontal)~~ ⇒ Actual Area 336.96 ft.^{2}
200.10 # of Rows of Panels in a sawtooth array = N.A. Rows
200.8, 200.9 Roof Area Required 13 ' 2 " High 26 ' 7 " Wide including insulation of approximately 2 inches wide around the perimeter (Provide additional space for attaching flashing to the roof as required.)

Section 200 — Step 2 — Solar System Flow Rate

Std. Design for space heating is 2 to 2½ CFM/ft.^{2} collector area. (See "Altitude and Temperature Corrections" on page 800.9.)

800.7, 200.2 (2 CFM/ft.^{2}) x (Actual Area = 336.96 ft.^{2}) = (674 CFM) ÷ (A.D.R. = .75) = $\boxed{900}$ CFM
200.6 (900 CFM) ÷ (336.96 ft^2) = 2.7 CFM/ft.^{2} and 2 panels in series = 0.42 " w.g. coll. pressure drop
200.7 Duct connections to the Collector = 14 " Diameter (inside diameter)
800.19 Main Duct size from collector array to Air Handler and Storage 14 " diameter or 12 " x 14 "

Section 400 — Step 3 — Heat Storage Unit

Std. Design is 1/2 to 3/4 ft.^{3} Rock per ft.^{2} collector.

400.5 800.7 (3/4 ft.^{3}/ft.^{2}) (Actual Area = 336.96 ft.^{2}) = (252.7 ft.^{3} Rock) x (0.037) = (9.3 yd^3) x (1.35) = 12.6 Tons
400.6 Select Storage Container Dimensions that provide Stratification.
800.7 Rock Depth = 5.5 ft. Floor Area of Container = (252.7 ft.^{3}) ÷ (Depth = 5.5 ft.) = 46 ft.^{2}
Heat Storage Unit Installation Instructions: Velocity = (900 CFM) ÷ (cross sectional floor area = 46 ft.^{2}) = 20 FPM
Rock Size of 3/4" to 1½", 5.5 ft. depth and 20 FPM ⇒ 0.16 " w.g. pressure drop
Inside Dimensions of Storage Unit for 46 ft.^{2} floor area = 5 ' 9 " Wide 8 ' 0 " Long
800.19 400.8 Ductwork connections to Storage Unit Top Plenum 8 " x 22 " Bottom Plenum 8 " x 44 " or equivalent round duct connections through the lid.

Section 300 — Step 4 — Solaron Air Handling Unit

[✓ (AU-0400)] ~~(AU-0500)~~ ~~(Custom Air Handler)~~

300.1 - 300.3 (See "Altitude and Temperature Corrections" on page 800.9.)
Air Handler Installation Instructions Pages 2, 7, 8, 9, 10, 11:
System Flow Rate 900 CFM System External Static Pressure 1.0 " w.g.
Motor RPM = 1160 Motor HP = 1/2 Solaron Drive Set VM50 Pulley AK56
Motorized Dampers (MD-1, MD-2) [✓ (AU-0400 = 14" x 14")] ~~(AU-0500 = 18" x 18")~~
See A.H.U. Install. Instructions Section 1200, page 10 for System Pressure Drop Criteria. ←

SYSTEM PRESSURE DROP

R.A. Grille	0.05 "
Filter	0.15
Backdraft damper	0.20
Collector	0.42
Ductwork	0.18
(Heat Storage)	—
Ext. Static Pressure =	1.0"

Section 300 — Step 5 — Dampers and Filter

300.4 Backdraft Damper Size At 900 CFM BD-1 and BD-2 = 20 " x 12 "
300.3 Motorized Damper MD-3 at 1000 CFM = 14 " x 14 " Size for Aux. CFM
800.8 Filter at ≅ 300 - 400 FPM : (1000 CFM) ÷ (350 FPM) = 2.86 ft.^{2} 20 " x 20 "
(Select Standard filter size.)

Section 800 — Step 6 — Auxiliary Heating System

800.18 Type [✓ (Elec.)] ~~(Oil) (Gas) (Heat Pump) (Other ______)~~ Number of Heating Zones [✓ one] (—)
600.3 - 600.14 Application ~~(Htg. only)~~ [✓ (Htg. and A.C.)] ~~(Htg. & Cont. Vent) (Night Set Back)~~ See A.H.U. Instal. Instru. p. 4 for Humidifiers and Electric Air Cleaners.
600.4 Aux. System Flow Rate 1000 CFM vs. Solar System Flow Rate from Step 2 Above 900 CFM
600.4 When the Two System flow rates are not equal, balance the system per Section 600. (SEE PAGE 600.4)

Section 500 — Step 7 — Solaron Controller

[✓ (Heating and Heating with Air Cond.)] ~~(Heat Pump)~~ ~~(Custom)~~

A few final words of caution on selecting pumps and blowers. Water flow rates should not exceed 8 feet per second in velocity or noise levels will be very high; however, it should be no less than 2 ft./second, in order that turbulent flow can optimize collector performance. Air flow should range from 200-400 ft./second in main trunks. DO NOT exceed the external static pressure rating of the blower. Otherwise, the motor will overheat.

Information on air patterns, spread, drop and throw is available from most manufacturers of registers, diffusers and grilles. Study this information, so that proper selection of these components can be made, if required.

SIZING, DESIGN AND RETROFIT

SYSTEM CONTROLS AND PROTECTIVE DEVICES

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

SYSTEM CONTROLS AND PROTECTIVE DEVICES

CONTROL FUNCTIONS

IMPORTANCE OF CONTROLS

The basic function of the control system is to ensure that a maximum amount of energy will be collected from a solar system to provide the required heating or cooling load to the structure. Controllers are extremely important in a solar heating and/or cooling system. There is a tendency for a great deal of concern for the efficiency of a collector and less concern for effectiveness of the control system. The net result is reduced effectiveness of a system. For example, one might spend a great deal of money for a good selective surface on the absorber of a collector to realize an improvement in efficiency of approximately 10 percent, whereas carefully designed controls could achieve that much or more for very little cost and effort.

The performance of solar energy systems is, in large part, a sensitive function of its control system. A typical control system consists of:

1. sensors that monitor the operating status of the solar energy system,
2. a means for making decisions (a microcomputer or some less sophisticated electronic device), and
3. a collection of actuators that can alter the operating state of the solar energy system.

Proper operation of a solar system depends on the design, installation and reliability of the control system. Improper control system operation can lead to costly problems. Some problems, such as collector freeze-up and pump cycling, are readily detected. Other collector related problems can result in severely diminished solar energy system performance but may not be easily detected without some means of quantitatively monitoring the operation of the system, as is done by the National Solar Date Network (NSDN).

It is convenient to group these control proplems into four major categories, as follows:

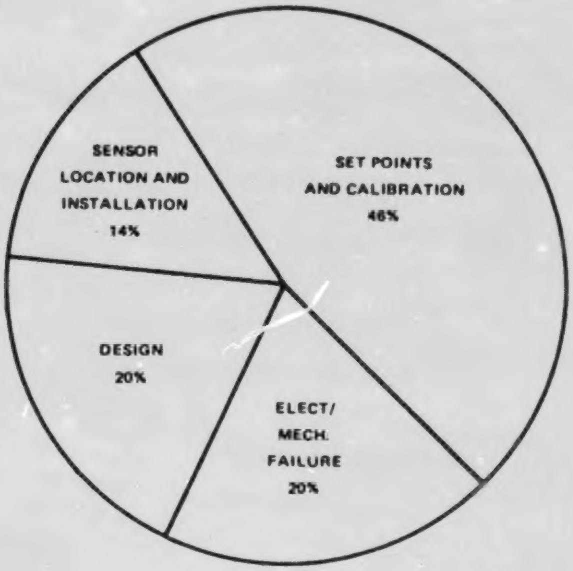
1. Design: This generally refers to an improperly matched control system with respect to a particular application. It also refers to such things as post installation modifications to a control system that may degrade system performance.

2. Sensor Installation and Location: This category covers controller problems arising from information which is incorrectly sensed. Such problems are frequently the result of improperly installing or locating a sensor (typically a temperature sensor) within the system.

3. Set Points and Calibration: This category deals with improper selection of set points or failure to calibrate a controller. Both affect the vital decision-making portion of the control system.

4. Component Failures: This is a broad category that covers electrical and mechanical failures of control system components. Included are failures such as malfunctioning dampers (mechanical linkage or electrical actuator motor), failed relays, valves and solenoids, as well as failure of various electronic components within the controller unit itself.

The relative percentage of overall occurrence of controller problems in each of these categories is shown in the figure below. The data is taken from reports of problems at 40 sites in the National Solar Date Network.



Distribution of NSDN Site
Solar System control Problems

Note that a significant majority of the problems (46%) are directly related to set points and calibration of the control sensors. Since set points are usually adjustable (many controllers have variable set points and calibration potentiometers), these problems often can be rectified at minimal cost to the solar consumer.

COLLECTOR CONTROL STRATEGY

Most controllers at present are on/off controllers, that is, they command a pump to be either on or off, depending upon certain conditions. The typical strategy with respect to controlling the flow of a transport medium in the collector loop is to start the collector pump whenever the collector fluid exit temperature is greater, by some set difference, than the tank temperature and to turn off the collector pump whenever the collector outlet temperature approaches the tank temperature. The temperature difference to start the flow is nominally set at 20° F and to stop the flow at 3° or 4° F.

As a specific example, suppose that the storage temperature is 120° F and the collector temperature is 50° F when the sun rises. The collector temperature will gradually increase, and when it reaches 140° F the controller will start the collector pump (assuming that the storage temperature is 120° F). Then in the afternoon as the sun begins to set, the collector temperature will begin to decrease. Suppose that the storage temperature has reached 150° F by 3:30 p.m. When the collector temperature decreases to 153° F the controller will stop the collector pump. The temperatures sensed by the sensors depend upon their location in the system.

Ratio of Temperature Differences

If the temperature sensor for the collector is located at a point such that the sensor is rapidly cooled by the transport medium, the result can be that the collector pumps will cycle on and off repeatedly. This cycling can also occur if the difference between the temperature to start and to stop the system is not properly selected. The ratio between the on-to-off temperature differences should be approximately five to seven. In the example given in the preceding paragraph, the starting temperature difference was 20° F and the stopping temperature difference was 3° F. The ratio is slightly less than seven. A larger value for this ratio will reduce the total energy collected by the system, while a smaller value will cause cycling.

Freezing Protection

Some controllers are designed to incorporate an aquastat to compare temperature of the transport medium with some preset temperature such as the freezing temperature of water. Then, if the temperature of the fluid in the collector approaches this preset temperature, the pumps are automatically started to circulate the fluid or to heat the fluid from storage in order to prevent freezing. This is not a recommended protection measure against freezing, because if there is a power failure during cold weather, the pumps will not operate and the collectors can freeze. It is preferred to use an antifreeze solution in the collector loop.

Two-Speed Pump

A two-speed pump may be considered as a possible way to regulate the temperature rise in the collector to improve collection efficiency. By changing to a slower flow rate during periods of low solar insolation, the system will collect heat at useful temperatures, whereas with a high flow rate, the temperature of the fluid at the collector outlet would be low and the control would stop the collector pump. When the solar radiation intensity is high, the flow rate can be increased. The fluid temperature would be reduced because of greater flow, and the collector will be more efficient.

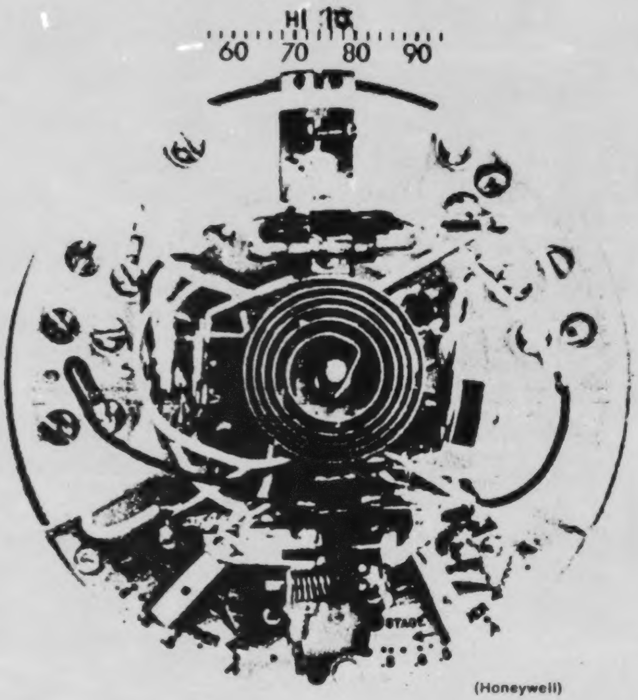
CONTROL SYSTEM HARDWARE

The solar system controls consist of power relays which switch electric valves and pumps in the liquid system, or blowers and dampers in the air system, and auxiliary heating units in both systems, in response to temperatures or temperature differences. Controls for solar systems fundamentally serve the same functions as conventional HVAC controls; however, there are more control functions in solar systems, and also there are "interlocks" which prevent undesirable or hazardous sequences of operation.

A solar system supplier should provide the required control hardware or at least specify it, along with explicit wiring instructions. Building a control system at the site should be avoided unless experience in this practice is available.

THERMOSTAT

A two-stage heat, indoor thermostat is recommended for residential solar heating systems, and a two-stage heat, one-stage cool type is recommended for solar heating and cooling systems. Variations will feature "on", "off", or "automatic" switches from heating to cooling or vice-versa to meet the need.



(Honeywell)

Fig. 8-1: Two-stage heating, single-stage cooling thermostat.

When cooling is required, the single-stage cooling provides indoor space temperature control. There is a dead band, which is a small range in temperature between start and stop signals given to the controller, which in turn controls the cooling system. The dead band for most thermostats is about 3° F. The heating operation is a bit more complex. Upon demand for heat, the first stage calls for the solar system to provide heat. If the building heat loss is greater than the solar system can provide, the temperature in the building will continue to drop to stage two and the auxiliary system will be called upon to provide heat. The auxiliary system can provide sufficient heat for the building by itself, or in combination with the solar system to raise the temperature in the room to the upper temperature limit of stage one which stops the heating system. The upper temperature dead band is nominally about 2° F.

The thermostat is the only control with which the occupant needs to be concerned. Once the occupant sets the winter comfort control level to, say, 68° F, and the summer comfort level to, say, 75° F (or other suitable temperatures), he should not have to select or adjust any other control in the heating and/or cooling system.

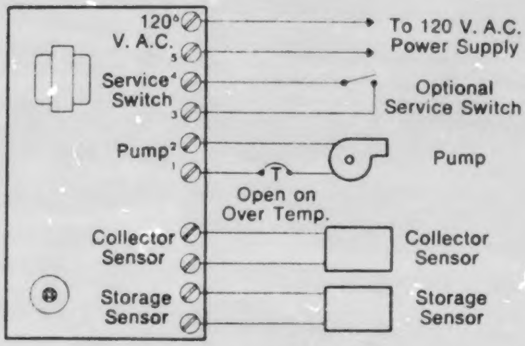
TEMPERATURE SENSORS/DIFFERENTIAL CONTROLLERS

An ordinary temperature controller is a thermostat that senses water or air temperature in various parts of the system to initiate control action; for example, stop or start fans and pumps and open or close damper valves. A differential temperature controller measures the difference between air or water temperatures at two or more locations in the system, as shown in Figure 8-2. There are any number of specialized differential controllers available to perform multi-control functions in a solar system. For example, one "proportional" differential controller varies pump speed as a function of the temperature difference between collector and storage tank. With only a 3° F difference between collector fluid temperature and storage temperature, the pump operates at slow speed. As the differential rises to 11° F, then the controller switches the pump to high speed.

These specialized differential controllers can also be "wired" to provide high temperature limit for storage and to provide a low temperature turn-on circuit for freeze protection. (The pump is started when outdoor temperature is about 37° F).

Outdoor Reset Controller

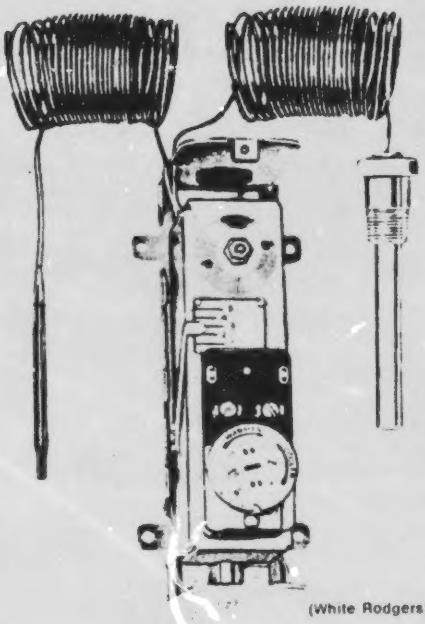
An outdoor reset controller senses outdoor air temperature. It is used to reset the control point of a temperature controller (either up or down) relative to changes in outdoor temperatures. (See Figure 8-3)


120 V. A.C.
6
5
Service Switch
4
3
Pump
2
1
Collector Sensor
Storage Sensor
To 120 V. A.C. Power Supply
Optional Service Switch
Pump
T
Open on Over Temp.
Collector Sensor
Storage Sensor

Typical Wiring Diagram

PENN
ELECTRONIC PANEL

(Penn Controls)



(White Rodgers)

Fig. 8-2: Differential Temperature Controller.

Figure 8-3: Reset Control.

Sensors

Every controller is made up of two basic components: a sensing device that measures temperature or pressure change, and a tranduces that converts that detection into electrical or mechanical action. Common sensors include the simple bimetal, remote bulb, and more recently, thermistor.

Bimetal Sensor. A bimetal sensor consists of two dissimilar metal strips bonded together (copper and iron are examples). They expand at different rates per degree rise in temperature, causing the metal strips to bend as illustrated in Figure 8-4.

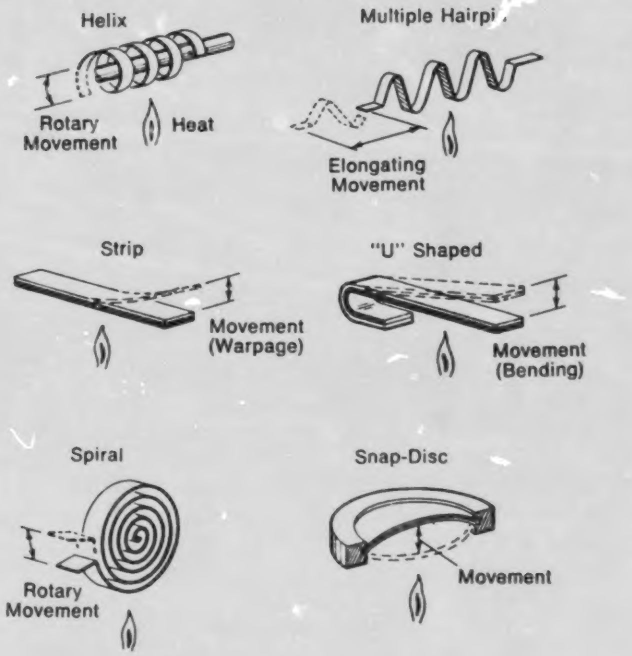

Helix
Rotary Movement
Heat
Multiple Hairpi
Elongating Movement
Strip
Movement (Warpage)
"U" Shaped
Movement (Bending)
Spiral
Rotary Movement
Snap-Disc
Movement

Fig. 8-4: Bimetal sensors

Bellows/Diaphram Sensor. This type of sensor uses a bulb to sense the temperature. The liquid in the bulb expands in the bellows or against the diaphram and causes the rod to move, which contacts the controller mechanism. An example of each type sensor is shown if Figure 8-5.

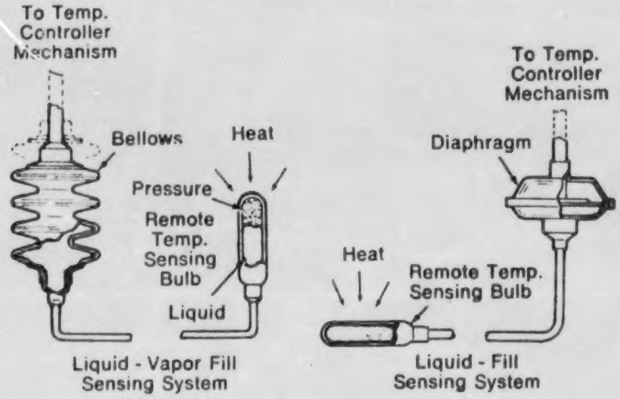
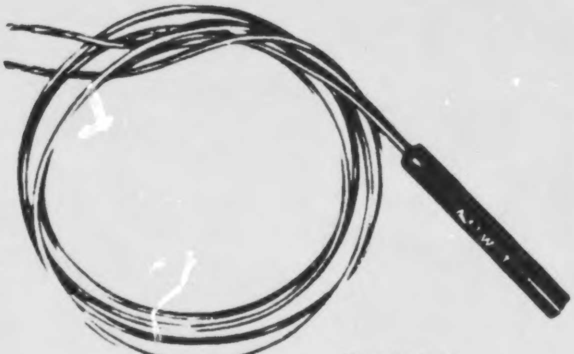

To Temp. Controller Mechanism
Bellows
Heat
Pressure
Remote Temp. Sensing Bulb
Liquid
Liquid - Vapor Fill Sensing System
To Temp. Controller Mechanism
Diaphragm
Heat
Remote Temp. Sensing Bulb
Liquid - Fill Sensing System

Fig. 8-5: Bellows/diaphragm sensors

Thermistor. A thermistor is a semi-conductor material with electrical resistance characteristics that vary with temperature. When connected to an electrical circuit, current flow varies in proportion to changes in the resistance of a thermistor exposed to changing air or liquid temperatures. Through the use of solid-state electronics, the current change is amplified to do work - open or close a relay, operate a solenoid, etc. Thermistor sensors can be located as far as 200 feet from the controller. Because of this feature, they are widely used to monitor solar systems. Ordinary 18 and 14 gauge wire can be used to connect sensors to a controller.

Thermistor sensors are relatively small components. They contain a nickel wire-wound element. The ability of a sensor to react to temperature change is called the temperature coefficient. For example, a sensor with a temperature coefficient of 3 ohms per degree Farenheit will increase or decrease its resistance, hence, current carrying capability, as ambient temperatures rise and fall. This change could signal the need for heat. The controller to which the sensor was connected would then activate the necessary heating system components.

The sensor in Figure 8-6 is encased in epoxy. It could be installed in an air collector circuit duct at the outlet of the collector to sense the stagnation temperature at which the circuit should be activated to prevent overheating the collector. It can also be placed in the heat storage unit at a location to sense the average temperature within the unit. In an air system, it can be inserted near the bottom among the rocks. For a liquid system, it would have to be placed in a bulb well (Figure 8-6a) so that the tank can remain watertight and not be drained if the thermistor requires service, and it will not make direct contact with the liquid. The corrosive effects of the liquid would damage the sensitivity of the sensor. When this sensor model is placed in a bulb, a thermal conducting compound is used to hold the sensor in the bulb so that the sensitivity to temperature change is not changed significantly. This compound is about the consistency of automobile lubricating grease.



(Penn Controls)

Fig. 8-6: Typical Thermistor Sensor

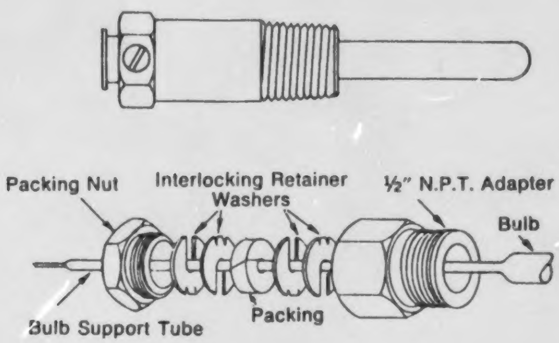


Fig. 8-6a: Typical Bulb Well.

There are other bulb well designs which can be used where heat sensing is desirable inside a pipe.

An alternature to using three components - the sensor, the bulb well, and the thermal compound - would be to use a preassembled component as shown in Figure 8-7.

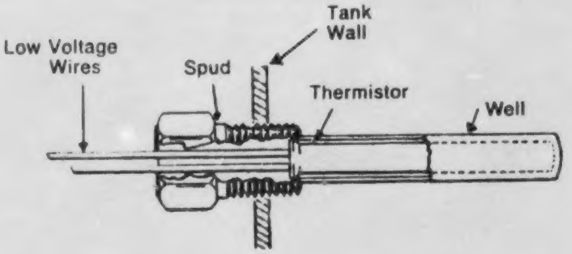

Tank
Wall
Low Voltage
Wires
Spud
Thermistor
Well

Fig. 8-7: Sensor installed through wall of tank.

This is a brass unit with the sensor sealed inside and has 1/2" pipe threads for mounting in a threaded hole in the heat storage tank or other control locations in the system.

It may be desirable to install a sensor in a piping system without the bulb well. If this is the case, a pipe Tee can be installed in the line and a sensor of the type shown in Figure 8-8 can be used.

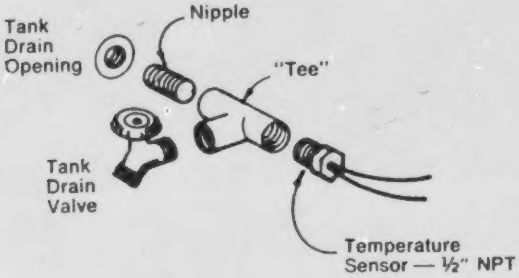

Tank
Drain
Opening
Nipple
"Tee"
Tank
Drain
Valve
Temperature
Sensor — ½" NPT

Fig. 8-8: Threaded Tank Sensor.

The sensor could be mounted in the side or one end of the Tee, depending on the installation specifications.

Placing a sensor near the bottom third of a liquid storage tank may be specified. For this type of installation, the sensor is screwed into a length of pipe. The length of the pipe is predetermined so that the sensor will function at the proper level in the liquid. The sensor screws into the end of the pipe with the wires extending up the pipe and out of the tank. The pipe is secured to the tank with a bushing as shown in Figure 8-9.

Another application of the sensor is shown in Figure 8-10. The sensor is mounted in the sheet metal duct and held in place with sheet metal screws in an air system.

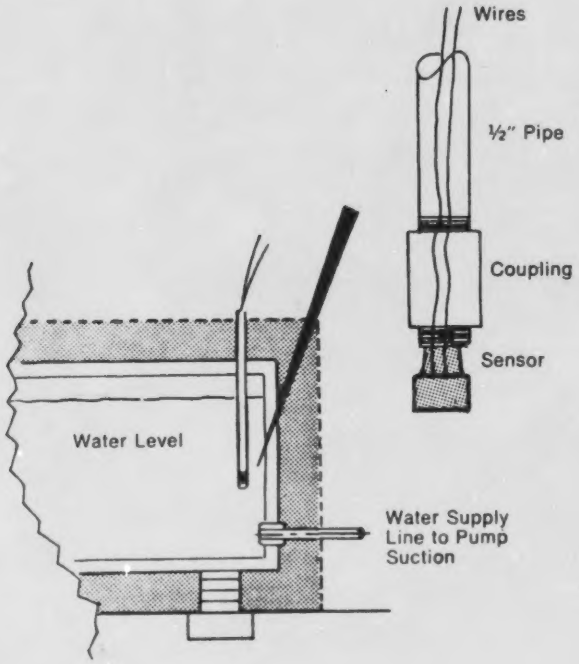

Wires
½" Pipe
Coupling
Sensor
Water Level
Water Supply
Line to Pump
Suction

Fig. 8-9: Sensor Fastened to end of Pipe Probe for Sensing Temperature in Deep Tanks.

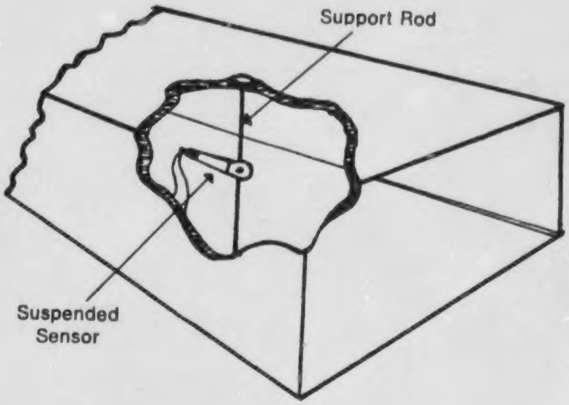
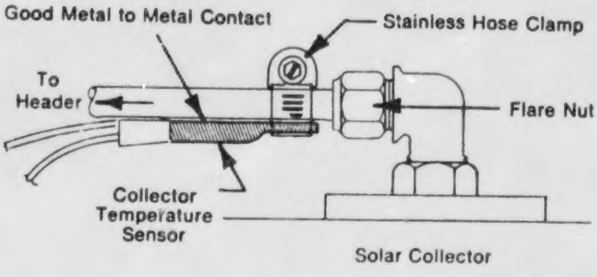

Support Rod
Suspended
Sensor

Fig. 8-10: Duct-Mounted Sensor.

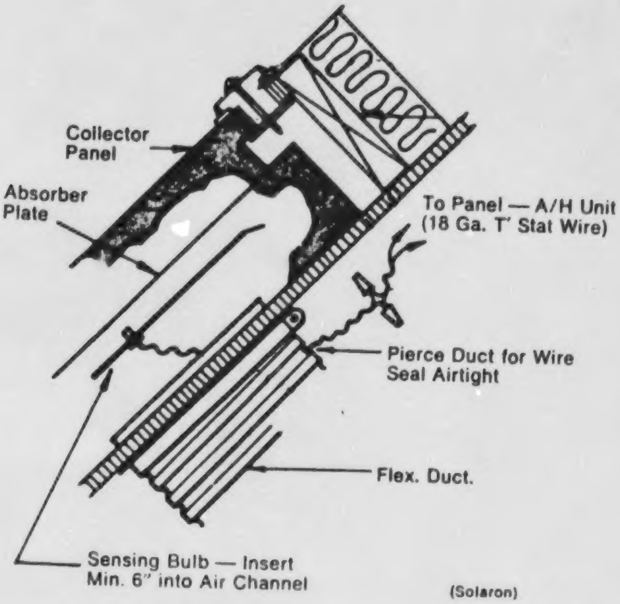
The sensor in Figure 8-11 is the same sensor as in Figure 8-6. However, it is in a more durable copper housing with a hole at the end to mount it more securely to a collector plate or elsewhere. A hose clamp can be used to secure this model on a pipe before the insulation is installed.

Figure 8-12 is typical of a sensor that would be mounted directly to the absorber plate in the collector to sense the absorber temperature indicating availability of solar energy at the collector. The sensor is attached with a sheet metal screw.


Good Metal to Metal Contact
Stainless Hose Clamp
To Header
Flare Nut
Collector Temperature Sensor
Solar Collector

(Lennox)

Fig. 8-11: Sensor Connected to Surface of Copper Discharge Line


Collector Panel
Absorber Plate
To Panel — A/H Unit (18 Ga. T' Stat Wire)
Pierce Duct for Wire Seal Airtight
Flex. Duct.
Sensing Bulb — Insert Min. 6" into Air Channel

(Solaron)

Fig. 8-12: Sensor Mounted in Air Collector.

Location

The locations of temperature sensors are not particularly critical, but there are some preferred locations. Temperature sensors are required to measure the air or liquid temperature as it exists from the collector, in the solar storage tank, or rock bed, and in the preheat water pan. The sensor in the conditioned space is the thermostat.

The sensor which measures the fluid temperature at the collector outlet can be located in the manifold which collects the fluid from the total array of collectors. It is preferred that the sensor be in contact with the fluid, but it is acceptable for the sensor to be in contact with the pipe, provided there is good thermal contact of the sensor with the pipe. If the sensor is attached to the outside of the outlet pipe, the sensor should be well insulated so that it does not lose the heat to the surroundings and register a low temperature. It is important to locate the sensor near the outlet so that it can register the fluid temperature when the sun is heating the collector but the fluid is not circulating. Sensors in the outlet manifold or absorber plate will register the increase in temperature, but the sensor located far from the manifold will not, and useful energy cannot then be collected. Wherever the sensor is located, the characteristics should be checked out when the system is put into operation.

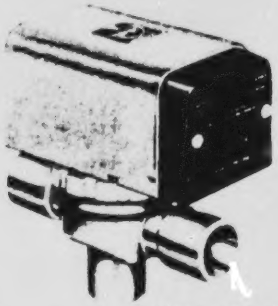
The sensor in the storage tank should be located near the bottom third inside the tank. When there is no fluid circulation, the temperature at the top of the tank will be slightly higher than the bottom, but while the fluid is in circulation, the fluid in the tank is uaually well mixed and the temperature will be uniform.

The location of the sensor in the preheat tank should be near the top one-third of the tank. If it were located near the bottom, the temperature at the top could be several degrees hotter. Also, when hot water is used in the household, cold water enters the preheat tank near the bottom. While the preheat tank would be thermally mixed when the pump is started, frequent cycling could result from the sensor registering locally cold water temperature. For an air system, the cycling is not particularly harmful because only one pump for the preheat cycle is involved. However, for the hydronic system, two pumps will be put into operation, and frequent cycling can be wasteful of electric energy. In both air and liquid systems, more heat would be lost than necessary from the pipes and heat exchangers because of frequent cycling.

The sensor in the pebble bed should be located at the bottom (or outlet) end of storage will determine if storage is "full".

Valve

Electrically powered solenoid valves are used to start and stop as well as divert fluid flow in a liquid system. They can be controlled by one or more thermostats. A typical valve is shown in Figure 8-13.



(White Rodgers)

Fig. 8-13: Motorized Valve.

Transformer

A transformer converts live voltage (usually 120V) to a lower voltage (typically 24 volts). This voltage drop is necessary because thermostats and many other control devices are designed for low voltage operation. A typical transformer and its schematic are shown in Figure 8-15.

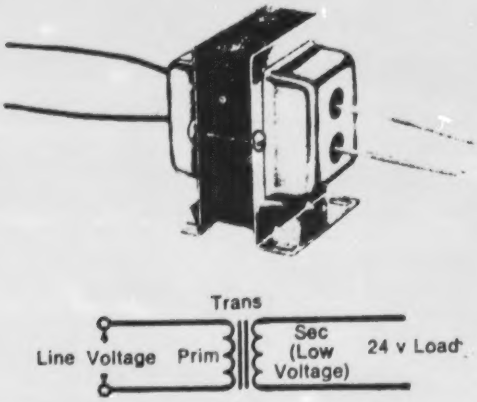

Trans
Line Voltage
Prim
Sec (Low Voltage)
24 v Load

Fig. 8-15: 24 Volt Transformer

Damper

Electrically powered multi-blade dampers are used to start and stop, as well as divert, fluid in an air system. They may be thermostatically controlled or opened and closed by the force of air from a blower. Figure 8-14 is an example of a motor-driven damper.

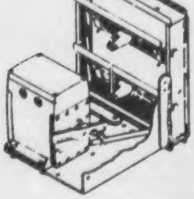


Fig. 8-14: Motorized Damper

Relay

A relay is an electromagnetic switch. This device (Figure 8-16) will often be operated by some low-voltage thermostat (24V) or other controller. When the relay is activated, it can cause a higher voltage load such as a 120 or 240V fan or pump to operate.

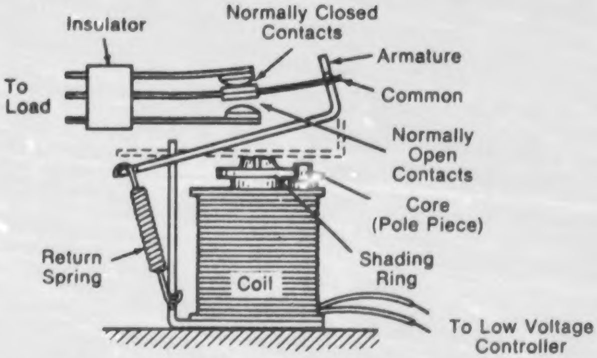

Insulator
Normally Closed Contacts
Armature
To Load
Common
Normally Open Contacts
Core (Pole Piece)
Shading Ring
Return Spring
Coil
To Low Voltage Controller

Fig. 8-16: Relay details.

A typical transformer-controller-relay-load control circuit is illustrated in Figure 8-17.

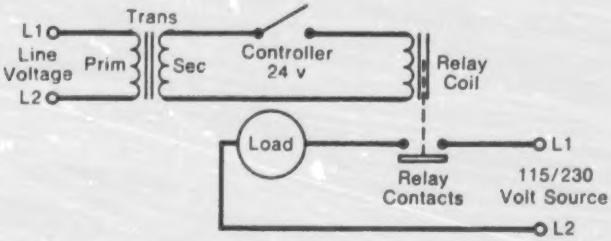

L1
Trans
Line Voltage
Prim
Sec
Controller 24 v
Relay Coil
L2
Load
L1
Relay Contacts
115/230 Volt Source
L2

Fig. 8-17: Transformer/relay Schematic

Some combination of the control devices explained above is necessary for the safe and automatic operation of a solar assisted heating system. However, there are some specialty items that are not electrically operated.

HYDRONIC SPECIALTY ITEMS

Hydronic specialties are non-electrical components that serve some control or system maintenance function. The following devices are representative of specialized components essential to the solar heating system.

Expansion Tank

In completely closed liquid systems, an expansion tank or "air cusion tank" is required to provide "room" for the expansion of heated water. For example, 50 gallons of water at 60° F, when heated to 200° F, would become 51.65 gallons at the elevated temperature. The cushion tank permits this expansion while also controlling the pressure in the system. Open or vented systems usually provide for the expansion within the storage tank as ullustrated in Figure 8-18.

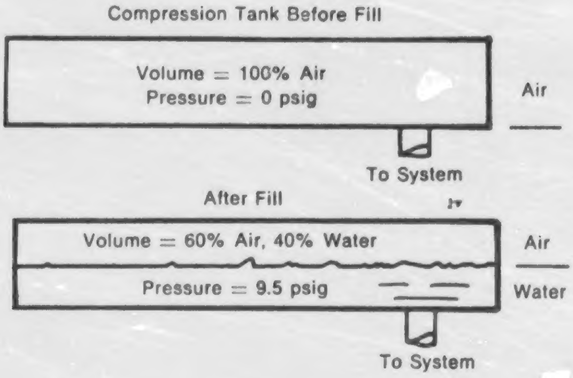
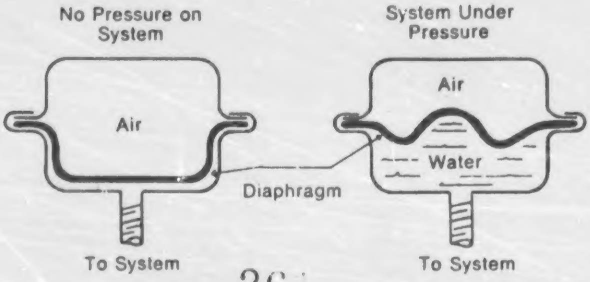

Compression Tank Before Fill
Volume = 100% Air
Pressure = 0 psig
Air
To System
After Fill
Volume = 60% Air, 40% Water
Pressure = 9.5 psig
Air
Water
To System

Fig. 8-18: Conventional Expansion Tank (above)
Diaphragm Design (below).


No Pressure on System
System Under Pressure
Air
Air
Water
Diaphragm
To System
To System

Air Vent

An air vent is placed at the highest point in the system (above the collector array). The devices illustrated in Figure 8-19 are of the float type, hydroscoptic disc, and manual design. They vent air from the system as it is being filled with liquid.

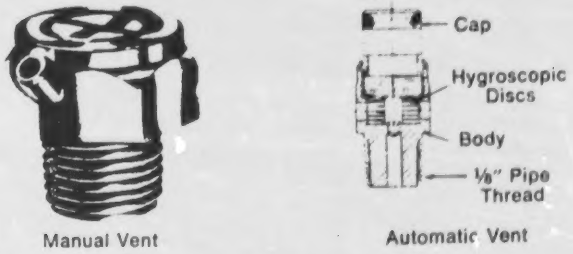
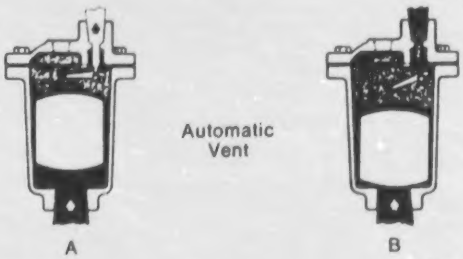

Manual Vent
Cap
Hygroscopic Discs
Body
⅛" Pipe Thread
Automatic Vent

Figure 8-19: AUTOMATIC air vent (above) operates by means of hygroscopic discs that when wet expand and seal venting ports. If air accumulates, discs dry and contract opening ports and venting air. (Dunham-Bush, Inc.) Float type vent (below) operates by means of a float that opens and closes a valve. In (A) water level is high and vent is closed. In (B) air has accumulated and water level is low causing float to open vent. (Armstrong Machine Works.) Manual vent (above) must be opened and closed by hand. (ITT Bell & Gossett.)


A
Automatic Vent
B

Check Valve

This type of valve limits fluid flow to one direction. There are a number of designs available for horizontal and vertical piping installation as illustrated in Figure 8-20. A flow control valve is a specialized check valve which prevents gravity circulation in the solar circuits when the pumps are off.

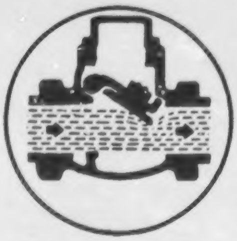
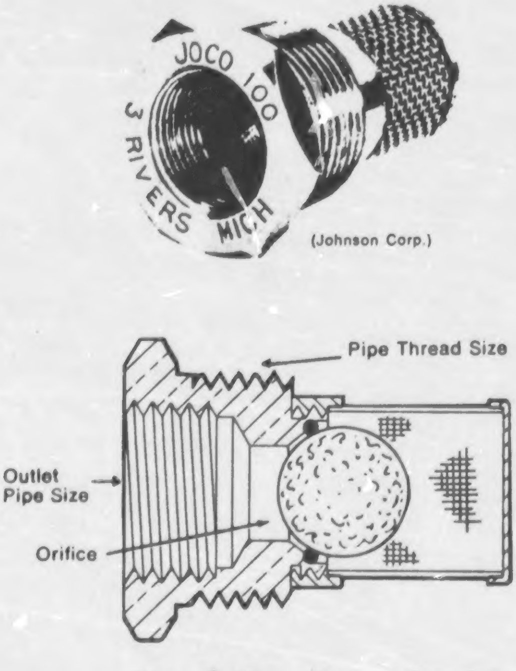


Fig. 8-20: Swing Check Valve - One of Several Designs

Vacuum Breaker

A vacuum breaker is sometimes used to relieve an unwanted vacuum condition in a drain down system. This valve, shown below in Figure 8-21, permits the system to drain efficiently by gravity, by admitting atmospheric pressure into the return piping.

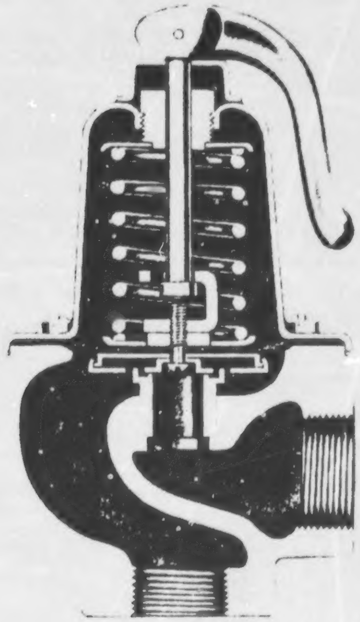


(Johnson Corp.)

Fig. 8-21: Vacuum Breaker

Pressure Relief Valve

A pressure relief valve, also known as a safety relief valve, is pressure operated to prevent excessive pressure in a closed system in the event of any malfunction. Figure 8-22 depicts one of these specialty items.

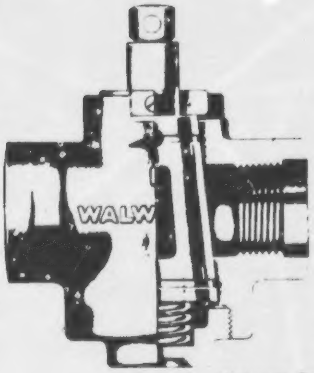


(ITT Bell & Gossett)

Fig. 8-22: Pressure Relief Valve

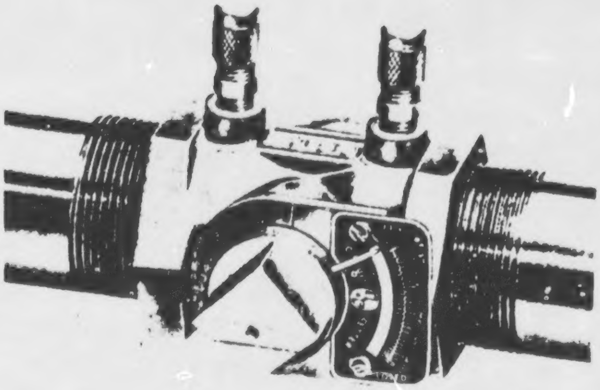
Balancing Valve

A balancing valve is a simple, inexpensive, square head cock. It is used to adjust flow rate in each circuit and especially through the collector array (see Figure 8-23). There are special, more deluxe, balancing valves used, as can be seen in Figure 8-24.



(Walworth Co.)

Fig. 8-23: Section view of plug or Square Head Cock Used to Adjust Flow in Multiple Circuit Systems.

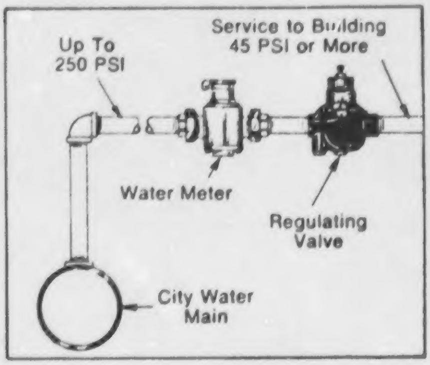


(ITT Bell & Gossett)

Fig. 8-24: Calibrated Balance Valve.

Pressure Reducing Valve

A pressure reducing valve is often used as part of the water supply system. Water service pressure that is too high for some of the heating system components can be reduced by using a valve such as illustrated in Figure 8-25.

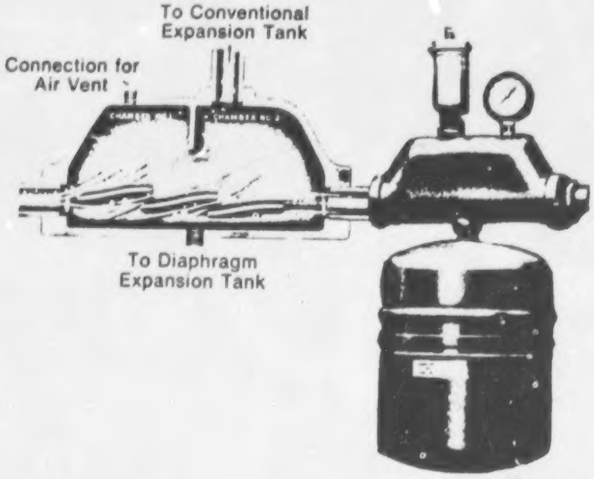

Service to Building
45 PSI or More
Up To
250 PSI
Water Meter
Regulating
Valve
City Water
Main

(Thrush)

Fig. 8-25: Pressure Reducing Valve

Air Eliminator

An air eliminator (Figure 8-26) is used in conjunction with an air cushion pressure tank to help purge a closed system of unwanted air.

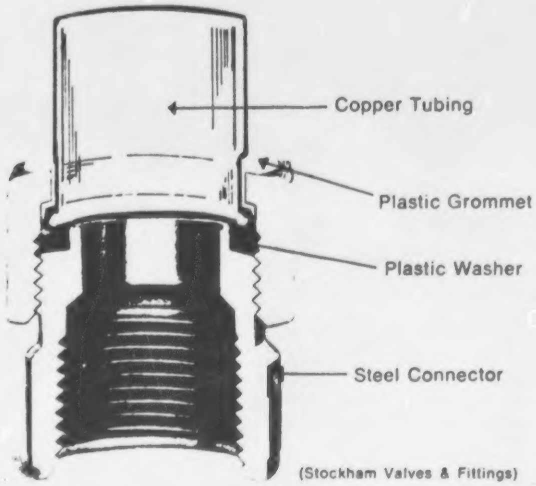

To Conventional
Expansion Tank
Connection for
Air Vent
To Diaphragm
Expansion Tank

(Taco)

Figure 8-26: Special Air Elimination Device
Connected to Diaphragm Tank.

Dielectric Union

Several dielectric unions may be needed in the system. Connecting fittings made of certain metals, may result in corrosion and restriction of liquid flow throughout the piping. Corrosion is most apt to occur when ferrous (iron) and non-ferrous (aluminum, brass copper) fittings and components are connected together. One way to eliminate the problem is with plastic fittings or to use dielectric unions as shown in Figure 8-27.



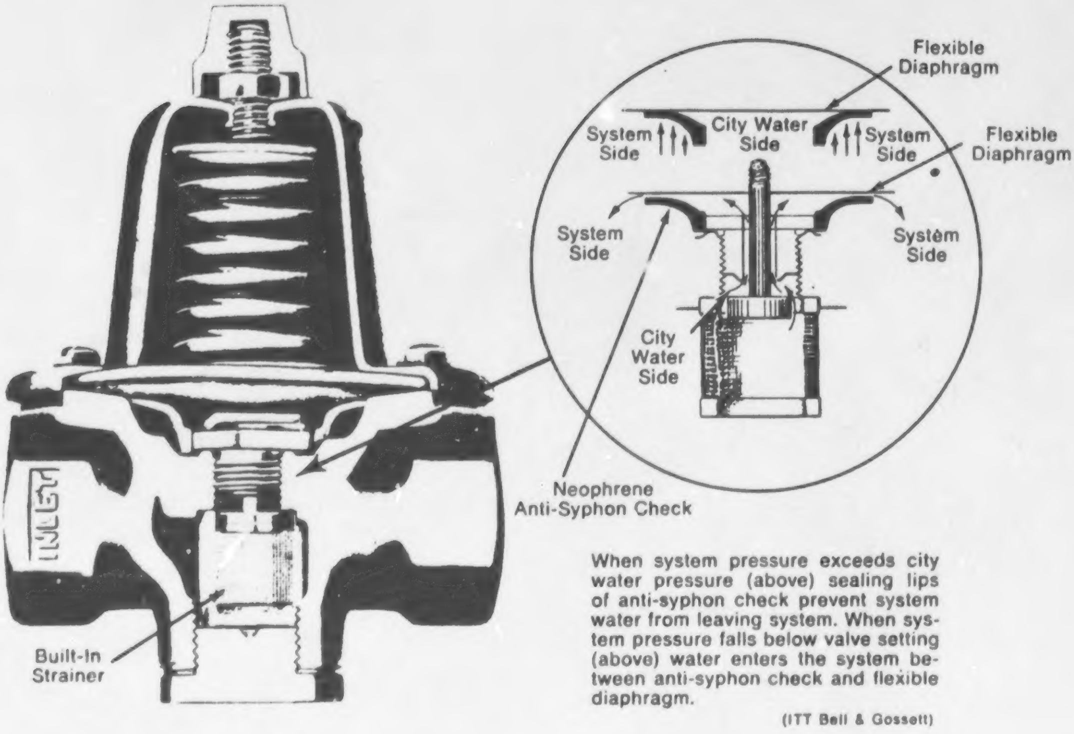
(Stockham Valves & Fittings)

Fig. 8-17: Dielectric Union.

Fill System

Filling the collector loop of a solar assisted heating system may be done automatically or manually. Manual filling practices must be followed if local building codes prohibit connection of a solar system containing a toxic fluid (anti-freeze) to city water service. An automatic fill system may be a convenience to the occupant but the (glycol) antifreeze may become diluted over a period of time. This would create a maintenance problem if freezing occurred. The manual fill system would include a pressure reducing valve, a check valve, and a manually operated globe valve.

An automatic system filling valve (figure 8-28) has a built-in strainer and check valve mechanism.


Flexible Diaphragm
System Side
City Water Side
System Side
Flexible Diaphragm
System Side
System Side
City Water Side
Neophrene Anti-Syphon Check
Built-In Strainer

When system pressure exceeds city water pressure (above) sealing lips of anti-syphon check prevent system water from leaving system. When system pressure falls below valve setting (above) water enters the system between anti-syphon check and flexible diaphragm.

(ITT Bell & Gossett)

Fig. 8-28: Automatic Fill Valve with Anti-Syphon Feature

Antifreeze-Filling Hook-Up

For a closed loop system that contains anti-freeze, a fill system for the glycol must be included. One example of such a system is in Figure 8-29.

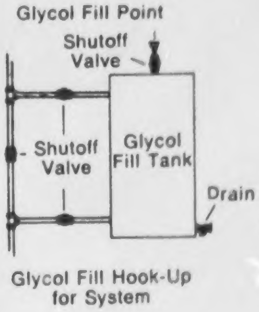

Glycol Fill Point
Shutoff Valve
Shutoff Valve
Glycol Fill Tank
Drain
Glycol Fill Hook-Up for System

Fig. 8-29: Glycol Fill System

Check Valve

A check valve (Fig. 8-30) restricts fluid flow to one direction. A check valve would be inserted in the system wherever there could be gravity back-flow or reverse flow caused by pressure.

B&G
FLO CONTROL VALVE
SIZE

Fig. 8-30: Check Valve

Gate Valve

A gate valve (Fig. 8-31) is a low-resistance manual shutoff valve. Gate valves are located at many points in the system piping, particularly on each of a major component. The gate valves are used to shut off fluid flow and isolate a component for servicing or replacement.

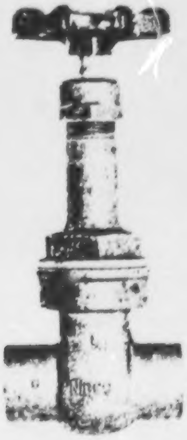


Fig. 8-31: Gate Valve

Three-Way Control Valve

The three-way valve (Fig. 8-32) is an electrically operated one-inlet, two-outlet valve. One outlet is normally open and the other outlet normally closed. In order to obtain the necessary flow volume in the collector piping loop, two identical valves are manifolded together. They are energized and de-energized in tandem. The valves are used to route fluids through the system.

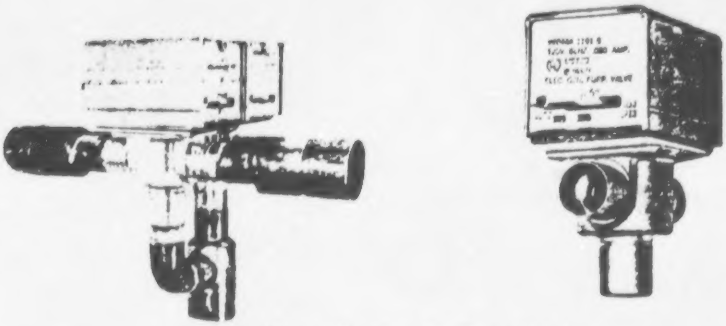


Fig. 8-32: Three-Way Control Valve

CONTROL PANELS

Usually a central control panel is convenient to consolidate the circuits and relays that provide the control functions. The panel would house the relays and provide for some adjustment of the temperature limits. It is best to acquire a control panel from the solar equipment manufacturer as a prewired unit to serve the system. All that needs to be done with a prewired control panel is to connect the temperature sensors, and motor, auxiliary, and valve and damper controls to the proper terminals in the control panel. The manufacturer will provide the necessary hook-up instructions. The power for the control panel will usually be household 115-volt single-phase A. C. line power.

TYPICAL CONTROL SUBSYSTEMS

AIR SYSTEM

A sketch of a typical configuration for an air system is shown in Figure 8-33, which is shown on the following pages. The temperature sensors are indicated by T_{ci}, T_{co}, T_S and T_E. T_{ci} measures the collector inlet temperature, T_{co} measures the storage temperature, and T_E measures the enclosure temperature. BD-1 and BD-2 represent backdraft dampers, D-1 and D-2 represent manual dampers, and MD-1 and MD-2 represent motorized dampers. The function of the controller is to operate the blowers and the motorized dampers to collect and distribute heat for the house. A flowchart of the control strategy is shown in Figure 8-34, which is shown on the following pages.

The first decision to be made by the controller concerns whether or not the house needs heat. This is determined by comparing the actual temperature with the desired temperature. If the temperature measured by the temperature sensor in the house indicates that the house does not need heat, then the controller must determine whether or not heat can be stored. This is determined by comparing the collector inlet temperature, T_{ci}, with the collector outlet

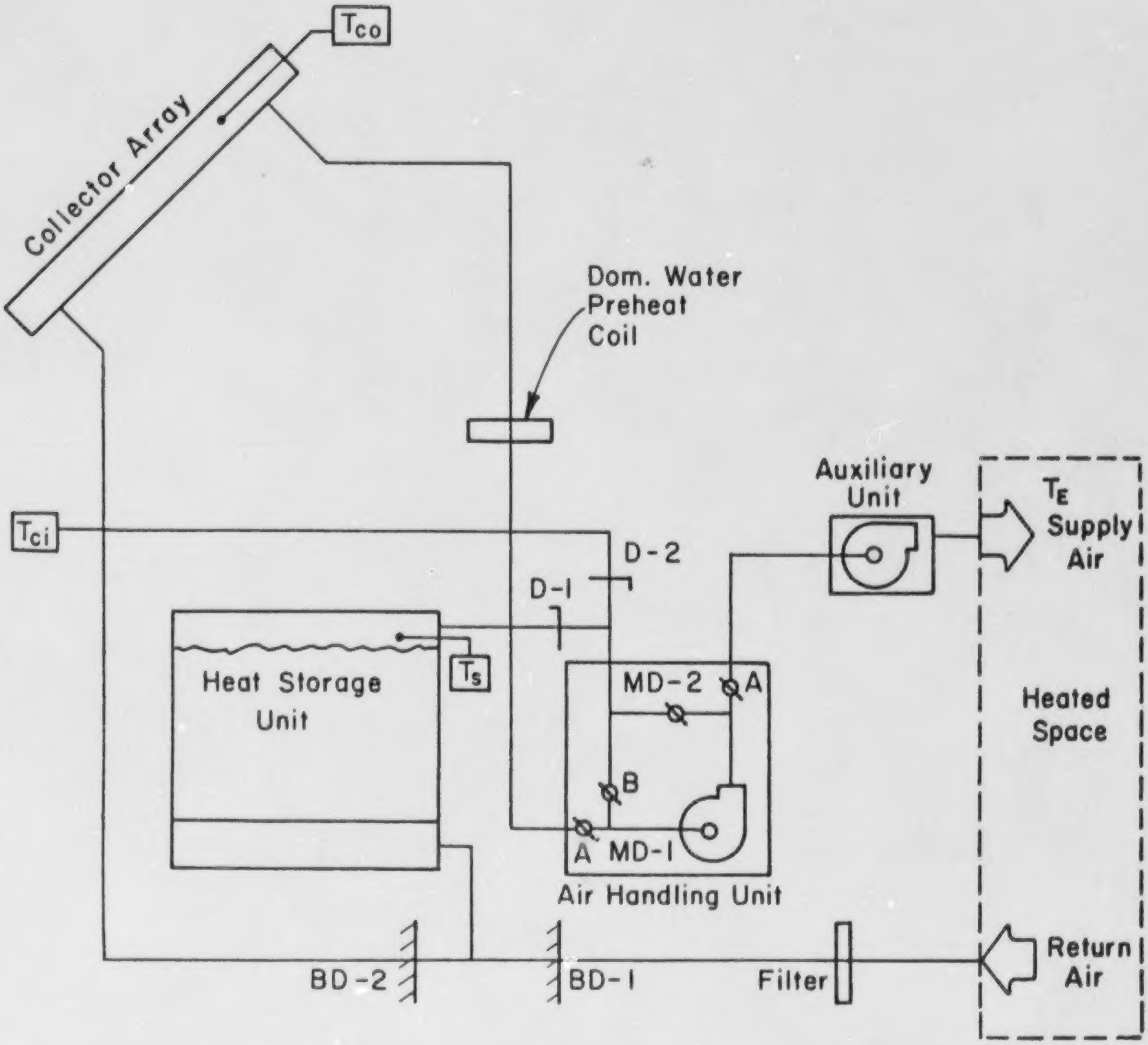

Tco
Collector Array
Dom. Water
Preheat
Coil
Tci
D-2
D-1
Auxiliary
Unit
TE
Supply
Air
Heat Storage
Unit
Ts
MD-2
A
B
A
MD-1
Air Handling Unit
Heated
Space
Return
Air
BD-2
BD-1
Filter

Figure 8-33: Schematic Diagram of Solar Air Heating System

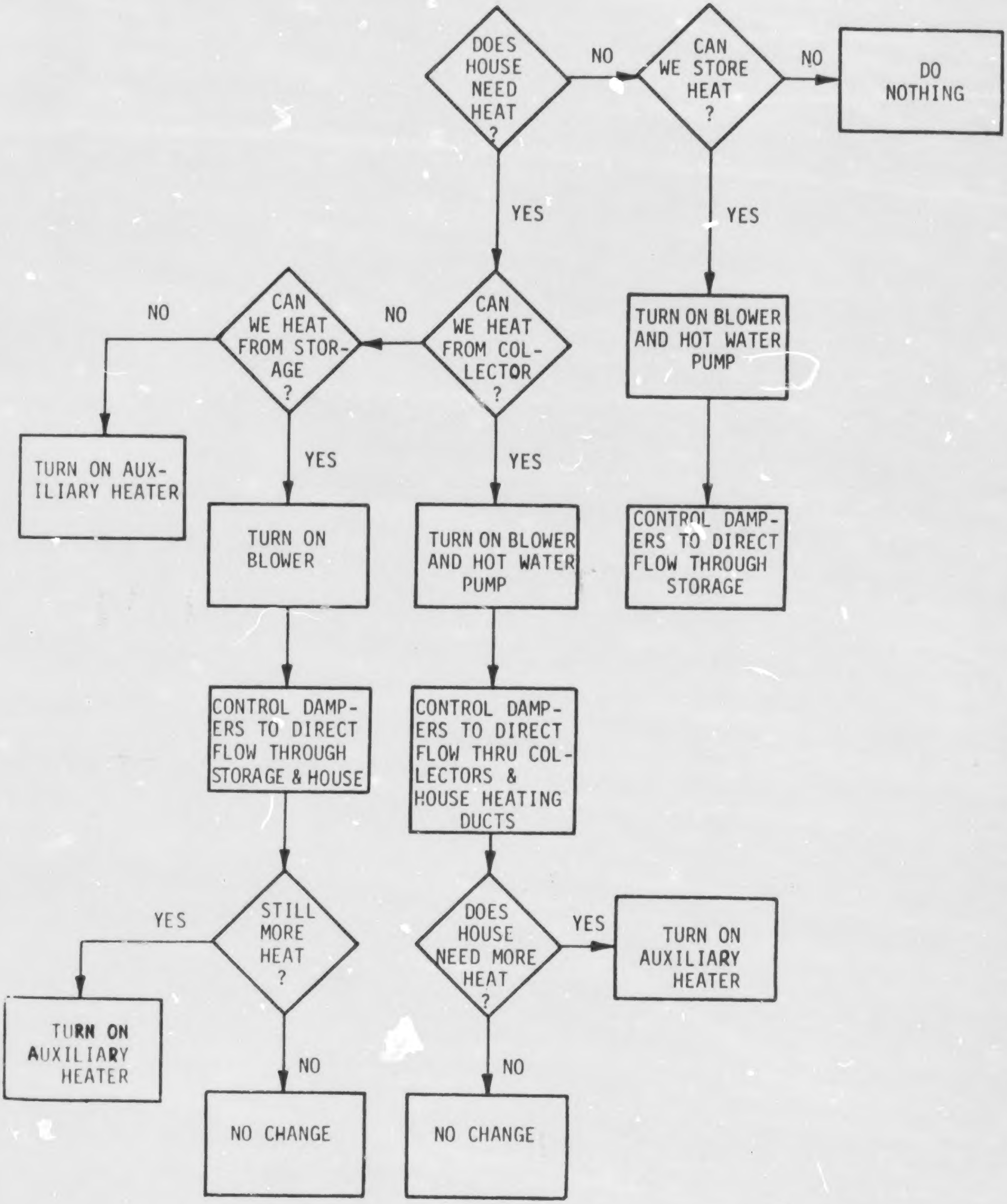

DOES HOUSE NEED HEAT ?
NO
CAN WE STORE HEAT ?
NO
DO NOTHING
YES
YES
TURN ON BLOWER AND HOT WATER PUMP
CONTROL DAMPERS TO DIRECT FLOW THROUGH STORAGE
CAN WE HEAT FROM COLLECTOR ?
NO
CAN WE HEAT FROM STORAGE ?
NO
TURN ON AUXILIARY HEATER
YES
TURN ON BLOWER
CONTROL DAMPERS TO DIRECT FLOW THROUGH STORAGE & HOUSE
STILL MORE HEAT ?
YES
TURN ON AUXILIARY HEATER
NO
NO CHANGE
YES
TURN ON BLOWER AND HOT WATER PUMP
CONTROL DAMPERS TO DIRECT FLOW THRU COLLECTORS & HOUSE HEATING DUCTS
DOES HOUSE NEED MORE HEAT ?
YES
TURN ON AUXILIARY HEATER
NO
NO CHANGE

Figure 8-34: Flow Chart of Control Strategy

outlet temperature, T_{co}. If T_{co} is not greater than T_{ci}, then obviously there is no heat being collected by the collector and we would not want to circulate the air through the collectors. If T_{co} exceeds T_{ci}, then there is useful energy available and the controller will turn on the blower and the hot water pump and control the dampers to direct the flow through the storage.

If the house requires heat, then the controller must determine whether that heat can be supplied from the collector or from storage. These determinations are again made by comparing temperatures. If T_{co} is not greater than T_{ci}, then obviously we cannot supply the heat from the collector; therefore, the next question to be asked is, can the heat be supplied from storage? This is determined by comparing the storage temperature, T_S, with the reference temperature, T_{SR}, which is nominally set at 100° F. If T_S is not greater than T_{SR}, then we can not heat from storage; if, however, T_S exceeds T_{SR}, then the controller will turn on the blower and control the dampers to direct the flow through the storage container and to the house by providing heat to the enclosure.

It is possible, of course, that the enclosure temperature may continue to decrease due to high heat losses from the house. If the enclosure temperature, T_E, drops below the reference temperature, T_{R2}, then it is clear that still more heat is required. In this case the controller will turn on the auxiliary heater.

If the controller determined that heat could be supplied from the collector, that is, T_{co} is greater than T_{ci}, then the controller will turn on the blower and the hot water pump and control the dampers to direct the flow through the collectors and then to the house heating ducts. It is still possible that this may not supply adequate heating to the house, and consequently the controller must compare the enclosure temperature with the second reference temperature to determine whether or not the auxiliary heater must be turned on.

The next step required in the design of a control system is to construct a truth table for the control strategy just described and represented in Figure 8-34. A truth table for this system is shown in Figure 8-35. The left-hand portion of the truth table shows the temperature comparisons. An entry of one in the truth table indicates that the statement is true, whereas a zero represents a situation where it is not true. For example, if T_E is less than T_{R1}, then a one would be entered in the first column in the truth table. The x's represent the "don't care" situation. The middle portion of the truth table shows the various operations. For example, a 1 in the B main column would indicate that the main blower is turned on, whereas a zero would indicate that the blower is turned off. Also, a 1 entry in the MD_1 column would indicate that port A on motorized damper number one is open and port B on motorized damper number one is closed. The right-hand portion of the truth table indicates the mode of operation. For example, consider the first line in the truth table. This line corresponds to the mode where the heat would be supplied from the storage. Suppose that T_E is less than T_{R1} but greater than T_{R2}; then we would enter a one in the first column and a zero in the second column. Suppose furthermore that T_{co} is not greater than T_{ci}; therefore we enter a zero in the third column. Suppose, however, that the storage temperature, T_S, is greater than the reference temperature, T_{SR}; we would want the controller to turn on the blowers and direct the flow through the storage to the house. This flow is directed by controlling motorized dampers one and two. When the blower comes on, it will draw air into the return air ducts, shown in Figure 8-33, through the filter, through backdraft damper number one, and then into the lower plenum on the storage unit. The air will then flow up through the storage unit, being heated in the process, and exit the storage unit at the top of the plenum. At MD_1 we must have B open and A closed, in order that the flow can reach the main blower. The air will exit the main blower and go through MD_2; at MD_2, we must have A open and B closed, in order that the flow of air be directed to the supply air ducts. Since it was assumed that T_E was greater than T_{R2}, it was not necessary that the auxiliary unit be turned on, and therefore a zero is entered in the gas column in the truth table. Also, since the collectors are not being operated, the hot water pump will not be turned on and, therefore, there is a zero entered in the HWP column of the

1	2	3	4	1+3	1	3	1	$1\cdot2 + 1\cdot\bar{3}\cdot\bar{4}$	3	
$T_E<T_{R1}$	$T_E<T_{R2}$	$T_{co}>T_{ci}$	$T_S>T_{SR}$	B_{main}	B_{aux}	MD_1	MD_2	GAS	HWP	MODE
1	0	0	1	1	1	0	1	0	0	Heating from storage
1	1	0	1	1	1	0	1	1	0	Heating from storage plus auxiliary
1	X	0	0	1	1	0	1	1	0	Heating from auxiliary
1	0	1	X	1	1	1	1	0	1	Heating from collector
1	1	1	X	1	1	1	1	1	1	Heating from collector plus auxiliary
0	X	1	X	1	0	1	0	0	1	Store Heat
0	X	0	X	0	0	X	X	0	0	Do Nothing

Figure 8-35: Truth Table for Control Strategy

truth table. The remaining modes of operation shown on the flow chart of the control strategy are illustrated in the truth table.

The next step required in the design of the control system is to select hardware to implement the logic shown on the truth table. The symbols at the top of the middle portion of the truth table represent the logic that is to be implemented; for example, above the B_{main} column we observe the notation 1 + 3. This indicates that the main blower is to be turned on whenever there is a 1 in column 1 or a 1 in column 3; similarly, the auxiliary blower is to be turned on whenever there is a 1 in column 1, port A on MD_1 is to be opened whenever there is a 1 in column 3, port A on MD_2 is to be opened whenever there is a 1 in column 1, and the gas is to be turned on whenever there is a 1 in columns 1 and 2 or a 1 in column 1 and a zero in columns 3 and 4.

A circuit diagram showing discrete components that may be used to implement the control logic just developed in shown in Figure 8-36, which will appear on the following pages. The comparators shown on the left-hand portion of the figure compare the various temperatures throughout the system. These signals are then sent through AND gates, OR gates, and inverters, in order to generate the signals that are sent to the motorized dampers, the blowers, and the auxiliary unit.

The solar system designer would not ordinarily concern himself with this level of detail, but would purchase a control unit that accomplishes the above-described tasks; it is important that the designer and the installer understand the functions of the control unit in order properly to conduct system check-out studies and ensure that the solar system operates as desired.

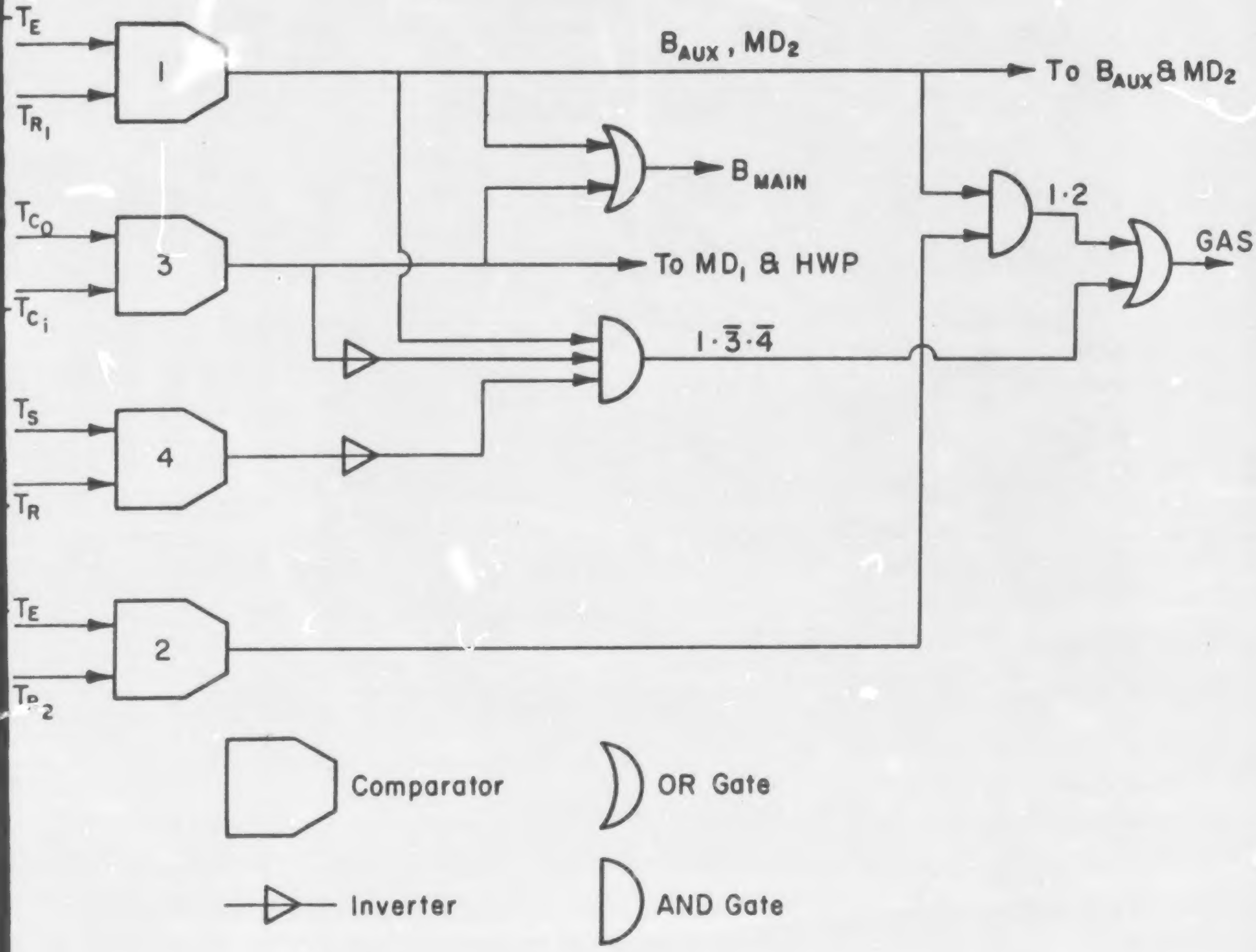

T_E
T_{R_1}
1
B_{AUX}, MD_2
To B_{AUX} & MD_2
B_{MAIN}
T_{C_O}
T_{C_i}
3
To MD_1 & HWP
$1 \cdot 2$
GAS
$1 \cdot \overline{3} \cdot \overline{4}$
T_S
T_R
4
T_E
T_{R_2}
2
Comparator
OR Gate
Inverter
AND Gate

Figure 8-36: Circuit Diagram.

HYDRONIC SYSTEM

A sketch of a representative hydronic installation is shown in Figure 8-37, which will appear in the following pages. The temperature sensors are indicated by S1, S2, S3, and S4. The signals from these temperature sensors are sent to the control panel in order to control the pumps, labeled P1 through P5, and the valves, labeled V1 and V2. Whenever the temperature recorded at S1 exceeds that of S2 by a preset amount, then the controller starts pumps P1 and P2 and heat is delivered to the storage tank. Also, whenever the temperature at S2 exceeds that at S4 by a preset amount, the controller will start pumps P4 and P5 and heat is supplied to the service hot water preheat tank. Finally, whenever thermostat S3 indicates that the house requires heat, pump P3 is started. The heat is supplied either from the storage tank or from the auxiliary boiler, depending upon the temperature of the water and the storage tank. If the storage tank temperature is not high enough, then the controller will set valves V1 and V2, so that the flow is through the auxiliary boiler and the auxiliary boiler is turned on.

The control for the pumps for the service hot water system operates in a manner similar to that previously described. The sensor S4 senses the temperature of the service hot water in the preheat tank. This is compared with the temperature of water in the storage tank and with a preset maximum value, for example, 150° F. When the temperature in the storage tank exceeds the temperature of the water in the service hot water preheat tank by approximately 10° F, then pumps P4 and P5 will be started, unless the temperature of the water in the preheat tank exceeds 150° F. We do not want the water in the preheat tank to exceed 150° F, in order to prevent scalding.

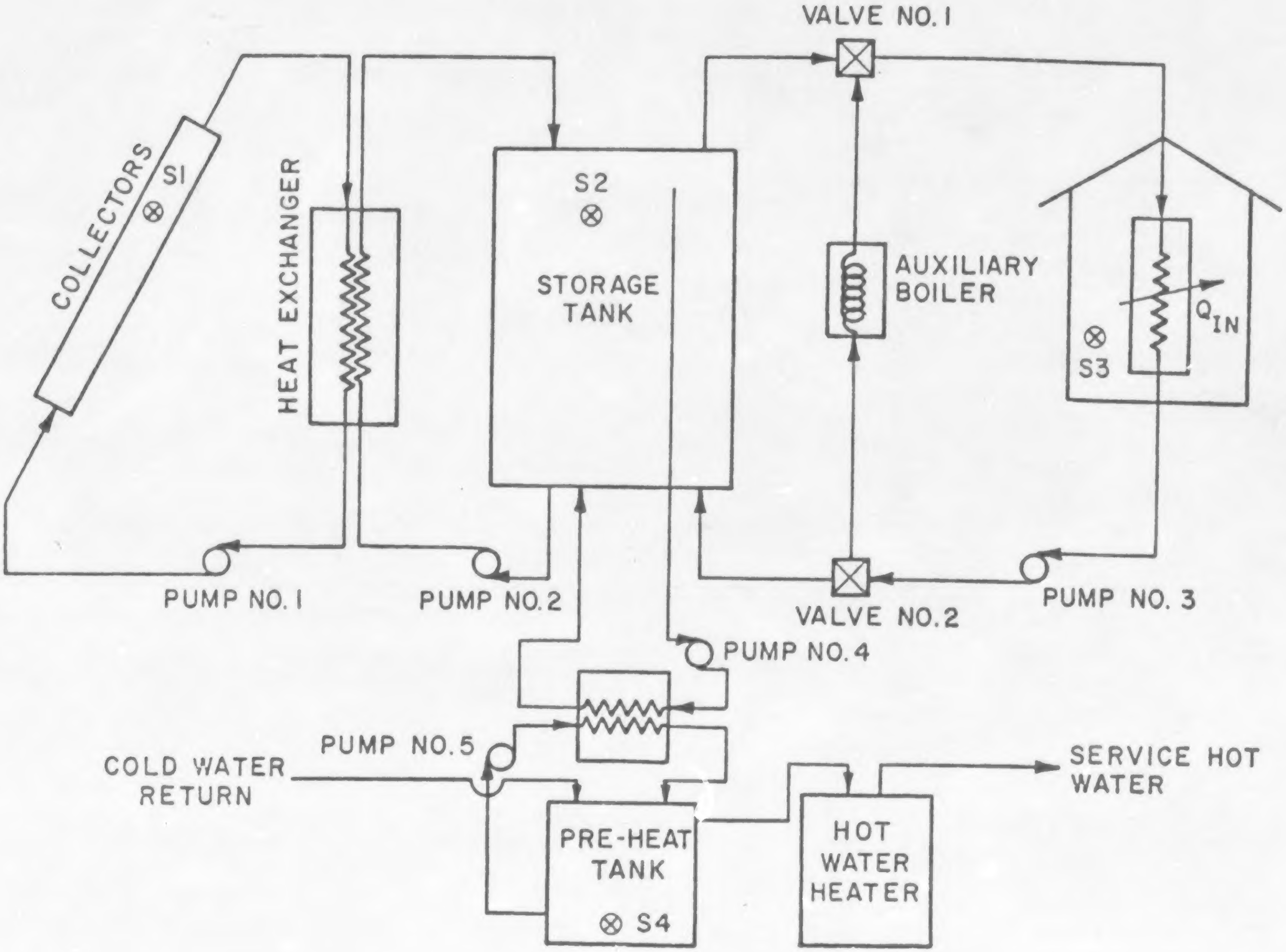
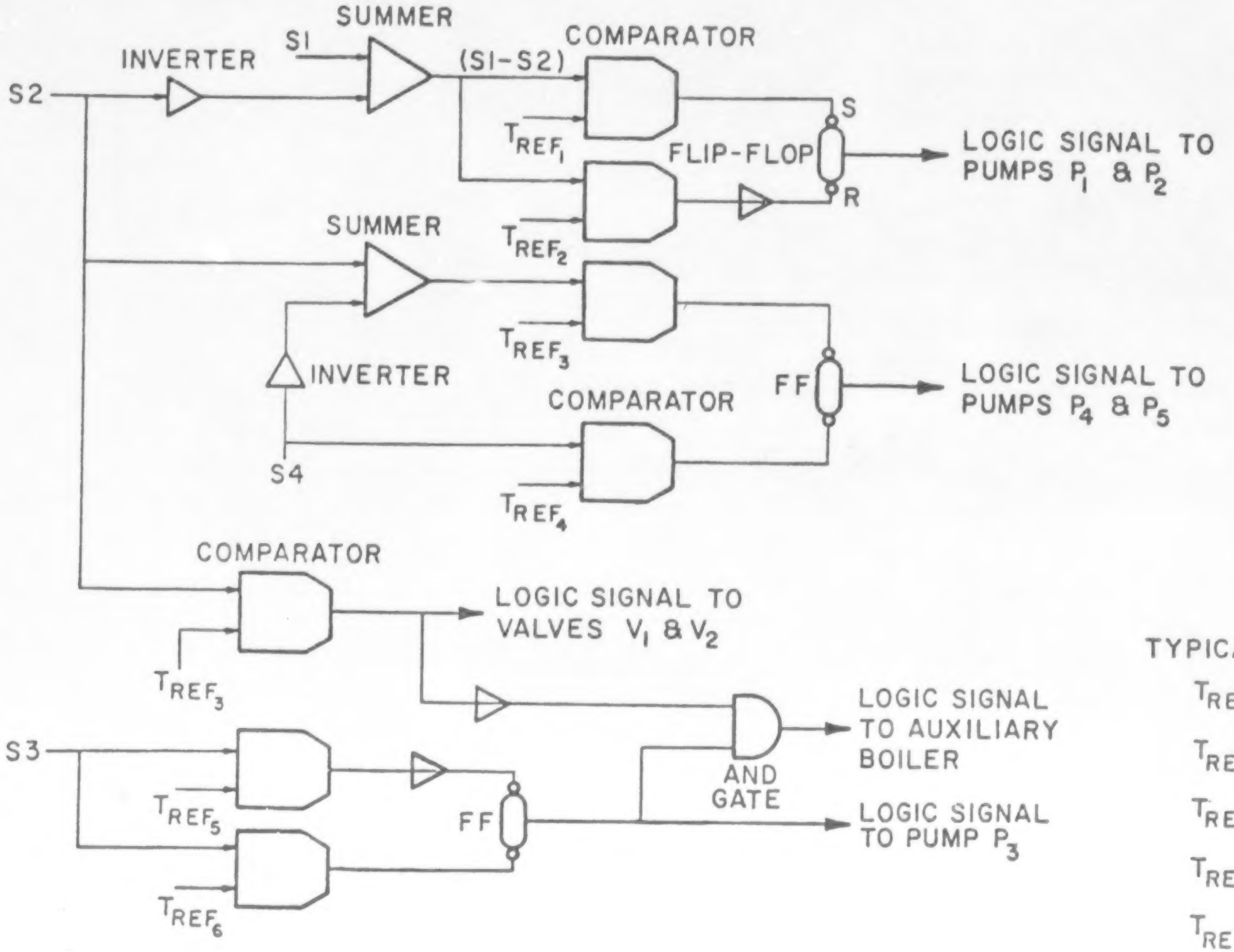

VALVE NO. 1
COLLECTORS
S1
HEAT EXCHANGER
S2
STORAGE TANK
AUXILIARY BOILER
S3
Q_{IN}
PUMP NO. 1
PUMP NO. 2
VALVE NO. 2
PUMP NO. 3
PUMP NO. 4
PUMP NO. 5
COLD WATER RETURN
SERVICE HOT WATER
PRE-HEAT TANK
S4
HOT WATER HEATER

Figure 8-27: Schematic Representation of a Typical Installation

CONTROL LOGIC, HYDRONIC SYSTEM

A schematic representation of a control logic for a typical installation is shown in Figure 8-38, which appears in the following pages. The system diagrammed in Figure 8-38 operates as follows: The signal from S1 is the temperature being measured at the collectors, while S2 is the temperature in the storage tank. The signals S1 and -S2 (-S2 is obtained by passing the signal S2 through an inverter) are combined in the Summer to produce the signal S1-S2. This is compared with $T_{ref\ 1}$ in the comparator in order to determine whether or not the temperature of the fluid coming out of the collector is high enough to justify turning on the pumps to collect and store the heat. As indicated earlier, this difference should be about 20° F. When this happens, the logic signal from the comparator is high (+ 5 volts) and this sets the flip-flop, which in turn will turn on pumps P1 and P2. These pumps must be turned off, however, whenever the difference between the collector output temperature and the storage temperature ceases to exceed approximately 4° F. In order to achieve this, the signal S1-S2 is compared with $T_{ref\ 2}$, then inverted, and then sent to the reset input of the flip-flop. The operation of the flip-flop device is such that when R is high, the flip-flop device will send a logic off-signal to pumps P1 and P2. Therefore, when the collector temperature minus the storage temperature is less than 4° F, the output of the comparator will be low, and consequently, the R input to the flip-flop will be high and this will turn off the pumps.

The control system for supplying heat to the house is shown on the bottom part of Figure 8-38. Thermostat S3 senses the temperature in the house, $T_{ref\ 5}$ represents the lower part of the dead band. and $T_{ref\ 6}$ represents the upper part of the dead band. Whenever the house temperature drops below $T_{ref\ 5}$, then pump P3 is started. This may be accomplished by comparing S3 with $T_{ref\ 5}$, as shown, then inverting the output of the comparator, and then sending the signal to the set input of the flip-flop. When the temperature in the house reaches the upper part of the dead band, the output of the second comparator will be high, which in turn will cause the flip-flop device to reset itself to turn off pump P3. Finally, the valves V1 and V2 are


SUMMER
S1
INVERTER
S2
(S1-S2)
COMPARATOR
S
T_{REF_1}
FLIP-FLOP
R
LOGIC SIGNAL TO PUMPS P_1 & P_2
T_{REF_2}
SUMMER
T_{REF_3}
INVERTER
FF
LOGIC SIGNAL TO PUMPS P_4 & P_5
COMPARATOR
S4
T_{REF_4}
COMPARATOR
LOGIC SIGNAL TO VALVES V_1 & V_2
T_{REF_3}
LOGIC SIGNAL TO AUXILIARY BOILER
S3
AND GATE
T_{REF_5}
FF
LOGIC SIGNAL TO PUMP P_3
T_{REF_6}

TYPICAL VALUES

T_{REF_1} = 20°F

T_{REF_2} = 4°

T_{REF_3} = 10°

T_{REF_4} = 150°

T_{REF_5} = 65°

T_{REF_6} = 70°

Figure 8-38: Schematic Diagram of the Control Logic for a Typical Installation

controlled by the output of the comparator that compares the temperature of the water in the storage tank with $T_{ref\ 3}$. If the storage tank temperature drops below $T_{ref\ 3}$, then valve V2 will direct the flow through the auxiliary boiler. The auxiliary boiler is to be turned on only if pump P3 is on and S2 is smaller than $T_{ref\ 3}$. This is accomplished by passing the two signals through the gate as shown in Figure 8-38.

CONTROL ACTUATORS

The pumps, blowers, valves, and dampers are referred to as the control actuators and produce the desired mechanical operation in response to the electrical control signals. Pumps and blowers are wired through manual switches from the control panel. The switch remains on; it is a safety feature and may even be required in the electrical codes. The switches are to be placed near the motors and not at the control panel.

Contral valves and dampers usually require some mechanical adjustment for proper setting. The best practice is to use spring-return two-position dampers or valves which are in the "normal" or most common position when unpowered.

SIZING, DESIGN AND RETROFIT

EQUIPMENT AND COMPONENT SPECIFICATIONS AND SELECTION

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

EQUIPMENT AND COMPONENT SPECIFICATIONS AND SELECTION

In addition to understanding the design and operation of solar heating systems, designers must be acquainted with several other aspects of solar heating. In order that intelligent selection of equipment can be made, knowledge of industry standards, equipment warranties, performance evaluation data, and related topics is necessary. If evaluations have been performed, their results should be available to the supplier and designer. The kinds of data required for such appraisal must be understood. The advantages and the disadvantages of the main system types for a specific application are particularly important. Knowledge of the types of hardware available, costs, and their compatibility with other components in the system is essential. Such items as safety, durability and operational reliability are additional criteria for equipment evaluation and selection.

Within this module, the main points enumerated above are addressed, and a guide to their consideration is presented. Because of (a) the newness of the solar equipment industry, (b) limited experience in the use of fully commercial systems in non-subsidized installations, (c) lack of criteria for system evaluation and certification, and (d) lack of information on durability, marketability, and other factors, much of the material outlined herein should be considered a guide rather than a set of specifications.

EQUIPMENT PERFORMANCE DATA

Most of the suppliers of solar heating system components provide technical data on their performance. Most of the collector data sheets contain information on solar heat collection efficiency at various temperatures and radiation levels. Some include information and instructions for sizing solar heating systems and installation procedures. At least two firms offer an extensive manual covering its products, instructions on their selection and sizing, and their assembly, installation, and servicing.

It should be recognized that some of the manufacturers' literature contains information which has not been verified by impartial analysis, and that the data may not be representative of performance under typical operating conditions. The user is advised to proceed with caution in applying manufacturers' performance figures that have not been independently verified.

Standardized procedures and instrumentation for testing solar equipment have been developed by the National Bureau of Standards (NBS) and are described in two reports:

1. "Method of Testing for Rating Solar Collectors Based on Thermal Performance", NBSIR-74-635. Hill and Kusuda, Center for Building Technology, NBS, December, 1974, Interim report prepared for the National Science Foundation.

2. "Method of Testing for Rating Thermal Storage Devices Based on Thermal Performance", NBSIR-74-634. Kelly and Hill, Center for Building Technology, NBS, March, 1975. Interim report prepared for the Energy Research and Development Administration.

Also, ASHRAE has established testing procedures in publication 93-77, which has become the most utilized procedure in recent months. The Solar Energy Industry Association (SEIA) is presently working on national standards to be adopted by manufacturers in an effort toward self-regulation that holds considerable promise for consumer protection and guaranteed performance.

Although the testing procedures described in these reports are not mandatory for the rating of equipment, they are being accepted by governmental purchasers of solar equipment.

Numerous solar collectors of the liquid heating type have been tested independently by the NASA-Lewis Research Center in Cleveland. Reports of their performance over a range of conditions are available and can be used as a guide to equipment selection. These test results may also be compared with the performance claimed by the manufacturers in their data sheets. Additional testing of liquid heating collectors is also in progress in several independent laboratories.

Facilities for testing and evaluation of complete solar heating systems are extremely limited. Colorado State University has three identical residential-type buildings in which various systems have been developed and evaluated. This program has produced information which can guide the choice of general system type, and has also yielded detailed operating data on specific systems. Many other installations, subsidized through the Department of Energy, have been evaluated and performance results are available through N.T.I.S. and NSDN.

SELECTION OF COMPONENTS AND SYSTEMS

Choice of equipment for solar heating involves a knowledge of the characteristics that are significant (and critical) and the advantages and disadvantages of each system type. Besides the information contained in this module, reference may be made to a helpful publication by SOLAR AGE MAGAZINE, "Solar Products Specification Guide".

Among the factors most important in equipment choice are the quality of materials and workmanship in the collector, controls, and fluid-handling equipment, the suitability of the materials and equipment to the application (involving such factors as durability, dependability, and safety), heat recovery efficiency over the range of operating conditions encountered, equipment cost, and installation cost. In addition to those factors mentioned above, the compatibility of the components in the total system, and how they affect the system's ability in performance of the application, is extremely important.

QUALITY OF MATERIALS AND WORKMANSHIP

Durable materials and high-quality workmanship are necessary for efficient, trouble-free operation of solar-heating systems. Visual inspection will often separate the good and poor equipment. Other criteria are records of satisfactory use in previous installations, compliance with minimum property standards, and recommendations from impartial specialists. With liquid systems, the collector, storage unit, heat exchangers, if used, and pumps and piping should be made of materials which are completely compatible with the liquids being used in order that corrosion will not prematurely damage or destroy the system or its components. The collector and other parts of the system must also be able to withstand the maximum and minimum temperatures to which they are exposed. The absorber plate in an efficient collector of the flat-plate type can reach temperatures above 350° F when fluid circulation is interrupted accidentally or purposely, and there should be no material in the collector not capable of withstanding no-flow temperatures for prolonged periods. Wood or other materials, which can outgas at these temperatures, should never be used in a solar collector. If inspection shows the presence of such materials, the collector is clearly unsuited to normal space heating applications.

SELECTION OF COLLECTOR

The efficiency of the collector in recovering solar energy in a heated fluid is the primary determinant of the size of collector required for supply of a particular fraction of the total heat requirements of a building. And, although this is an important criterion for collection selection, installed cost per unit area is equally significant. Assuming two styles of collectors have equal durability, the one having the greater heat delivery per dollar of first cost is the superior choice, regardless of the efficiency and the costs themselves. In other words, an increase of a few percentage points in efficiency, which might be achieved by doubling the cost per square foot, is not advantageous.

The corrosiveness of water in contact with aluminum or steel, in the presence of air, is a factor which must be considered in the design and use of water-heating solar collectors. Galvanic corrosion (in the presence of other metals) of aluminum in water must be avoided by suitable non-conducting connections in the system. Pitting corrosion of aluminum in the presence of slight metallic impurities, as well as dissolved oxygen and impurities in the water, may result in early failure of the aluminum tubes, particularly if thin-walled. Breakdown of anti-freeze solutions (ethylene glycol, for example) to acidic compounds can accelerate corrosive attack and must be avoided by suitable preventive maintenance.

Steel is less subject to attack than aluminum, but precautions must nevertheless be taken. The probable life of a steel collector is greater than that of an aluminum collector having the same tube thickness. Periodic draining and filling with air must, however, be avoided. Copper, at least for tubes, appears to be the most durable and dependable material. The only disadvantage is its substantially higher cost. A plate-type copper collector requires an outlay roughly three dollars per square foot in excess of that for aluminum. At the retail level, this difference could be as much as five to six dollars per square foot in selling price.

With any of the metals used for water-heating collectors, corrosion inhibitors can be added to the solution (whether freeze-protected or not) thereby substantially extending the life of the equipment. The inhibitor itself, however, must be maintained at suitable concentration by periodically checking and adding when necessary.

CRITERIA AND STANDARDS

Although no performance criteria or standards for solar heating equipment have yet been established, several such efforts are being made. Among the active organizations are the American Society for Testing and Materials (ASTM), the American National Standards Institute (ANSI), the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE), the Sheet Metal and Air-Conditioning Contractors' National Association (SMACNA), and various government bureaus, including the National Bureau of Standards (NBS), the Department of Housing and Urban Development (HUD), and the Department of Energy (DOE). Also, many states have testing centers and bureaus of standards which influence standards.

A committee of the ASTM and ANSI organizations is actively engaged in formulating standards for solar heating equipment. No results have been publicly released, but criteria or guidelines may be expected.

An important project of the National Bureau of Standards is the formulation of performance criteria, which solar heating and cooling equipment should be expected to meet. Two of the results of this project are the reports, "Interim Performance Criteria of Commercial and Solar Heating and Combined Heating/Cooling Systems and Facilities", NASA 98M-10001, 28 February, 1975 (prepared by NBS), and "Interim Performance Criteria for Solar Heating and Combined Heating/Cooling Systems and Dwellings", HUD, 1 January, 1975 (prepared by NBS for HUD). These publications contain information on the characteristics of solar systems and components which are important in the selection of equipment. No requirements are outlined, in terms of quantitative performance, but the equipment is expected to perform at the level which the manufacturer or supplier specifies. In addition to the criteria themselves, the reports describe methods for measuring the performance of collectors and heat storage units.

Another government effort along these lines has resulted in the release of "Intermediate Minimum Property Standards Supplement for Solar Heating and Domestic Hot Water Systems", prepared by the National Bureau of Standards for the Department of Housing and Urban Development. In conformance with other HUD documents of this type, the specifications outlined are those which solar heating equipment will have to meet if federal funds, such as FHA home loans, are used in financing the structure or its components. As with the "interim performance standards" developed by NBS, the solar heating and cooling standards in the HUD document are directed mainly to safety, durability, reliability, and such factors rather than to the specific efficiency of heat supply or other quantitative criteria. The equipment is required to perform according to the manufacturer's claims.

The work undertaken by SMACNA has been directed toward standards for installation workmanship in solar heating systems. Such factors as the quality of the plumbing, sheetmetal work, and electrical work are considered.

Standards for testing solar equipment have been the subject of work at the National Bureau of Standards for over two years. A useful report of part of this investigation is "Development of Proposed Standards for Testing Solar Collectors and Thermal Storage Devices", NBS Technical Note 899, issued February, 1976.

Another document related to standards and criteria, prepared at the Center for Building Technology of the National Bureau of Standards for the Energy Research and Development Administration, Division of Solar Energy, is "Thermal Data Requirements and Performance Evaluation Procedures for the National Solar Heating and Cooling Demonstration Program". This manual provides detailed information and directions for measuring and evaluating the performance of solar heating and cooling systems.

WARRANTIES

The types of warranties offered by manufacturers of solar heating equipment vary considerably. At the present time, if a supplier provides any warranty, it is of the "limited" type. Under its terms, the equipment is warranted to be free of defects in materials and workmanship, and that if such defects are found within a certain period of time after initial use, correction or replacement will be made without cost to the user. The warranty period varies with different manufacturers, and must be an integral part of the selection process.

There appear to be no manufacturers' guarantees as to thermal efficiency or heat delivery capability of solar equipment. Although manufacturers are providing that type of information in their sales literature, they are not guaranteeing the performance in the field. To a certain degree, this omission is due to the inability of the manufacturer to control the quality of the installation. In addition, manufacturers supplying only certain components of a system, such as the collector, cannot be assured that the other components in the system are correctly selected or integrated with their own product. Thus, inferior performance might well be due to factors other than those controlled by the collector manufacturer. A performance warranty would thus be difficult to establish and maintain. Hopefully, recent efforts in performance warranties toward industry standards will bring about a change of attitude.

Still another problem in providing a meaningful performance warranty is the great variation in climate encountered and the practical difficulty in accurately measuring the output of the installed equipment. Instrumentation is usually not provided, so measurement of performance is likely to be an expensive investigation by an experienced engineer. Disputes, litigation, and other problems would be inevitable. (Legalities, codes and warranties are explored in depth in the course, "Economics, Codes, Legalities and Consumerism.")

The many negative aspects, previously presented, make the selection process appear to be impossible; however, this is not the case. Be very careful in making your choice. Selecting equipment manufactured by reputable companies greatly reduces the risk.

Skill in decision-making, as to which model number of a particular component, is essential. Careful consideration and examination of the specification sheets of the equipment reveal that (those) factor(s) that make the choice clear.

Example I: Examine the spec sheets on the Libbey-Owens-Ford Co. Liquid Collectors, Models 120 and 121 (Figures 10-1 and 10-2). The suggested list price, net area, flow rates, application and general construction are the same for both panels. However, the performance curve indicates one is superior due to absorber coating - black chrome. At this point, the decision would be easier depending upon application and cost per Btu's delivered.

Solar Products Specifications Guide

SUNPANEL, MODEL 120

Libbey-Owens-Ford Co.
1701 East Broadway
Toledo, OH 43605

Lloyd E. Bastian (419) 247-4355

TECHNICAL SPECIFICATIONS

Type: Liquid

Model: 120

Applications: Domestic hot water, space heating

Collector dimensions - gross; net aperture area: 36 x 84 x 4-3/4 in; 19.25 ft^2

Performance test data:
  F_R ($\tau\alpha$): 0.75
  F_R (U_L): 1.47 (Btu/hr-ft^2-°F)
  Based on net area and 14.7 lb/hr/ft^2 flowrate (DSET Labs)

Glazing: Single tempered glass, float or low iron, replaceable

Recommended flowrate and pressure drop: 0.5 gpm per collector; 0.4 psi glycol/water at 140°F

Absorber panel:
  Material: Copper
  Thickness: 0.016 in
  Surface treatment: Flat black paint

Tube material and spacing: 3/8 in copper on 3-3/4 in centers

Headers: 5/8 in copper, 1 in also available

Insulation, back and edge: 3 in fiberglass

Manifold details: Inlet, lower left; outlet, upper right on side; also 1 in internal

Recommended heat transfer fluid: Glycol/water or silicone fluids

Case material: 0.077 in extruded aluminum, mill finish

Sealants and adhesives: UV-inhibited silicone

PRODUCT DESCRIPTION

An all copper absorber collector for use with water, aqueous glycol solutions or silicone fluids. The extruded aluminum case has welded corners, no screws or bolts, and is weather sealed; 3 in. of fiberglass insulation.

Features: Integral handles on case function as hold-down brackets when collector is installed.

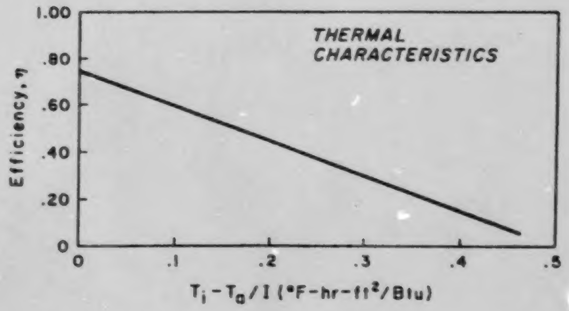
Options: Clear float or low iron glass, selective or non-selective absorber coating.

Installation Requirements: None.

Maintenance Requirements: None.

Guarantee/Warranty: 1-yr limited warranty.

Suggested List Price: $12 to $14./ft^2.



(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-1

SUNPANEL, MODEL 120

TECHNICAL SPECIFICATIONS (cont.)

Spacer materials and desiccants: Aluminum, fiberglass and rubber; no desiccants, vented design

Temperature limitations on entire collector: 400°F, no flow

Method of installation to roof: Wind-tested mounting system

Collector weight: 90 lb filled

Limitations: None

Guarantee--thermal degradation/chemical breakdown of components due to temperature: None

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-1, continued

Solar Products Specifications Guide

SUNPANEL, MODEL 121

Libbey-Owens-Ford Co.
1701 East Broadway
Toledo, OH 43605

Lloyd E. Bastian (419) 247-4355

TECHNICAL SPECIFICATIONS

Type: Liquid

Model: Model 121

Applications: Domestic hot water, space heating

Collector dimensions - gross; net aperture area: 36 x 84 x 4-3/4 in; 19.25 ft^2

Performance test data:
F_R ($\tau\alpha$): 0.71
F_R (U_L): 0.86 (Btu/hr-ft^2-°F)
Based on net area and 14.7 lb/hr/ft^2 flowrate (DSET Labs)

Glazing: Single, tempered glass, float or low iron, replaceable

Recommended flowrate and pressure drop: 0.5 gpm per collector; 0.4 psi with glycol/water at 140°F

Absorber panel:
Material: Copper
Thickness: 0.016 in
Surface treatment: Black chrome

Tube material and spacing: 3/8 in copper on 3-3/4 in centers

Headers: 5/8 in copper, 1 in also available

Insulation, back and edge: 3 in fiberglass

Manifold details: Inlet, lower left; outlet, upper right on side; also 1 in internal

Recommended heat transfer fluid: Glycol/water or silicone fluids

Case material: 0.077 in extruded aluminum, mill finish

Sealants and adhesives: UV-inhibited silicone

PRODUCT DESCRIPTION

An all copper absorber collector for use with water, aqueous glycol solutions or silicone fluids. The extruded aluminum case has welded corners, no screws or bolts, and is weather sealed; 3 in. of fiberglass insulation.

Features: Integral handles on case function as hold-down brackets when collector is installed.

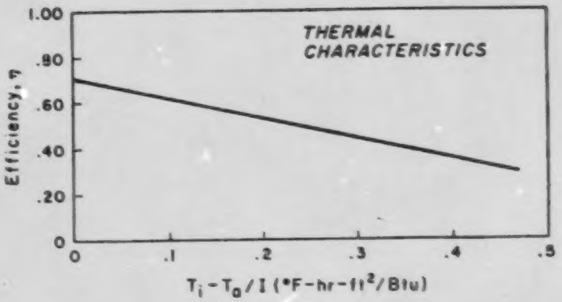
Options: Clear float or low iron glass, selective or non-selective absorber coating.

Installation Requirements: None.

Maintenance Requirements: None.

Guarantee/Warranty: 1-yr limited warranty.

Suggested List Price: \$12 to \$14/ft^2.



(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-2

SUNPANEL, MODEL 121

TECHNICAL SPECIFICATIONS (cont.)

Spacer materials and desiccants: Aluminum, fiberglass and rubber; no desiccants, vented design

Temperature limitations on entire collector: 400°F, no flow

Method of installation to roof: Wind-tested mounting system

Collector weight: 90 lb filled

Limitations: None

Guarantee--thermal degradation/chemical breakdown of components due to temperature: None

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-2, continued

Example II: Suppose a differential controller was needed, and the choice had been narrowed down to two models, made by different companies. For capabilities and reliability of performance, either one would be acceptable (Figures 10-3 and 10-4). Which one would be the best choice?

Examination of the spec sheets reveals that drain down freeze protection may be the deciding factor for one, while adjustable high limit may swing the choice to the other.

Example III: In selecting a pump for a domestic hot water application, the choice has been narrowed down to two manufacturers' model groups (Figures 10-5 and 10-6). Examine these two spec sheets and notice that there is considerable difference in head capacity, delivery rate, power requirements and operational characteristics. There is also considerable difference in initial cost. If either one will do the task, which one would you choose?

Solar Products Specifications Guide

SOLAR HOT WATER CONTROLLER 80-171

Solar Control Corp.
5721 Arapahoe
Boulder, CO 80303

Liz Quinn (303) 449-9180

TECHNICAL SPECIFICATIONS

Type: Differential thermostat

Dimensions: 4 x 4-1/2 x 5-1/4 in

Application: Solar domestic hot water heating systems

Power input: 115 vac, 3 watts

Switching capacity: 10 amps

Operation: Solid state logic, relay output

UL listed: Yes

Standard differentials available: 20°F on, 4°F off

User adjustable: Yes; on, off, auto switch

Sensors supplied: No; available separately from factory

Sensitivity: ±2°F over range

Operating range: Controller chassis -40°F to 140°F, sensors -40°F to 300°F

PRODUCT DESCRIPTION

A solid state differential thermostat capable of fully controlling a solar domestic water heating system. The unit is designed for a life-time of maintenance free service and incorporates freeze and boil protect circuitry.

Features: On-off-auto switch, LED "on" indicating light.

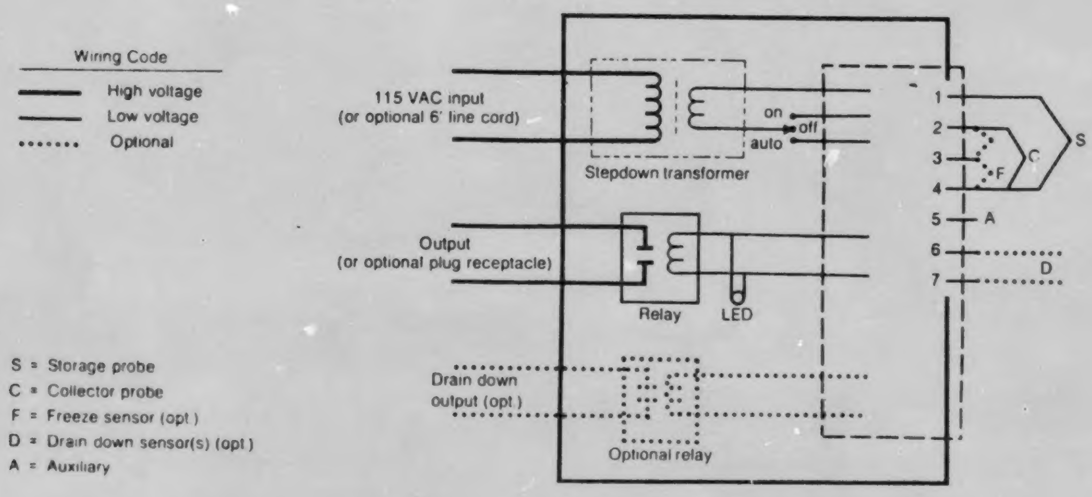
Options: Drain-down freeze protection, 6 ft. line cord and plug receptacle.

Installation Requirements: Mounts on 4 in. x 4 in. J box, auxiliary relay required for motors greater than 1/4 hp.

Maintenance Requirements: None.

Guarantee/Warranty: 1-yr. warranty.

Suggested List Price: $65.


Wiring Code
High voltage
Low voltage
Optional
115 VAC input
(or optional 6' line cord)
Stepdown transformer
on
off
auto
Output
(or optional plug receptacle)
Relay
LED
Drain down
output (opt.)
Optional relay
1
2
3
4
5
6
7
S
C
F
A
D
S = Storage probe
C = Collector probe
F = Freeze sensor (opt.)
D = Drain down sensor(s) (opt.)
A = Auxiliary

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-3

Solar Products Specifications Guide

C-30 SUN SWITCH

Independent Energy, Inc.
P.O. Box 732, 42 Ladd St.
East Greenwich, RI 02818

L. Tinkoff (401) 884-6990

TECHNICAL SPECIFICATIONS

Type: Differential

Dimensions: 4-9/16 x 5 x 2-1/2 in

Application: DHW, space and pool heating

Power input: 117 vac

Switching capacity: 1/3 hp

Operation: Solid state circuitry, mechanical relay

UL listed: Yes

Standard differentials available: 20°F on, 5°F off; 8°F on, 3°F off

User adjustable: High limit is adjustable from 110 to 212°F+

Sensors supplied: No

Sensitivity: ±2°F

Operating range: 32 to 212°F

PRODUCT DESCRIPTION

A differential temperature control for solar domestic hot water. Applicable to most air and hydronic systems. Features quick attach, pump mount Fit Kits.

Features: Adjustable storage high temperature limit, freeze protection, LED indicators.

Options: Ac line cord, outlet, 1/2 hp relay, 230 vac input, output NO.

Installation Requirements: None.

Maintenance Requirements: None.

Guarantee/Warranty: 1-yr. limited warranty.

Suggested List Price: C-30, less options, $65.85.

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-4

V-S-296

Solar Products Specifications Guide

CLOSED SYSTEM CIRCULATORS

Grundfos Pumps Corp.
2555 Clovis Ave.
Clovis, CA 93612

Ole Mathiasen (209) 299-9741

TECHNICAL SPECIFICATIONS

Type: Centrifugal

Horsepower: See table

Maximum flowrate: See curves

Maximum head: See curves

Construction/materials: Aluminum housing, cast iron volute, stainless steel impeller

Maximum continuous operating temperature: 230°F fluid temp

Self-lubricating: Yes

Self-priming: No

Recommended heat transfer fluids: Water, 50% glycol solutions

Limitations: Manufacturer must approve use of other fluids for warranty to be honored

PRODUCT DESCRIPTION

These Grundfos domestic circulators are single stage, direct drive, centrifugal pumps, designed primarily for closed system applications. They can be operated up to system pressures of 142 psi with fluid temperatures of 230°F and corresponding ambient temperatures of 68°F. 50% by volume mixtures of water and ethylene or propylene glycol solutions may be pumped. Check with the manufacturer for information regarding the suitability of other fluids.

Features: None.

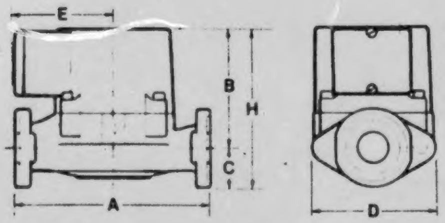
Options: 3/4 to 1-1/2 in flange sets, special models available for high heat applications and for use with hydrocarbon fluids.

Installation Requirements: Pump shaft whould not fall below horizontal plane.

Maintenance Requirements: None.

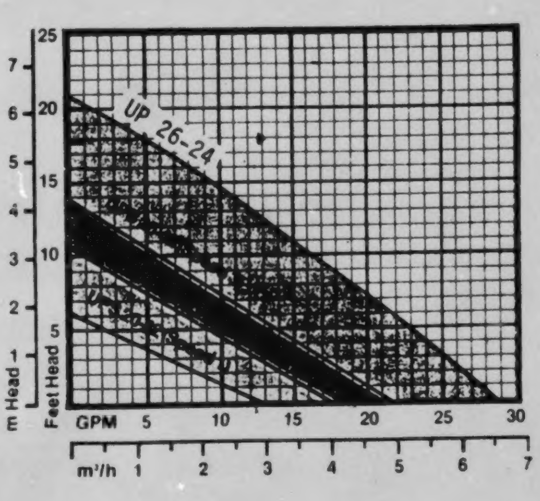
Guarantee/Warranty: 18-mo. conditional warranty.

Suggested List Price: $130. to $145.



Model	A	B	C	D	E	H
UPS 20-42F 3-Speed	6 1/2	4 3/16	1 5/16	4 3/16	3 9/16	5 1/2
UP 26-64F	6 1/2	5 1/16	1 5/16	4 3/16	3 9/16	6 3/8

Model	Speed	Hp	Watts	Volts	Amps	RPM	Capacitor
UPS 20-42 3-Speed	3	1/20	95	115	0.85	2620	10 MF / 180V
	2	1/32	70		0.60	2300	
	1	1/64	50		0.42	1800	
UP 26-64 High Velocity	1	1/12	185	115	1.65	3200	8 MF / 180V



(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-5

Solar Products Specifications Guide

SOLAR CIRCULATOR 008F

Taco, Inc.
1160 Cranston St.
Cranston, RI 02920

Kurt L. Mumpton (401) 942-8000

TECHNICAL SPECIFICATIONS

Type: Circulator 008F

Horsepower: 1/25

Maximum flow rate: 13 gpm

Maximum head: 16 ft

Construction/materials: Cast iron (pump housing), non-ferrous impeller; also available in bronze construction

Maximum continuous operating temperature: 240°F

Self-lubricating: Yes

Self-priming: Yes

Recommended heat transfer fluids: Water, water with propylene glycol or ethylene glycol

(See Taco Circulator 006B - Chart and drawings)

PRODUCT DESCRIPTION

The first self-lubricated circulator specifically designed for use in closed systems, this 1/25 hp pump delivers 13 gpm and a maximum head of 16 ft.

Features: The pump housing is cast iron, the impeller is non-ferrous; all moving parts are housed in a patented, stainless-steel cartridge.

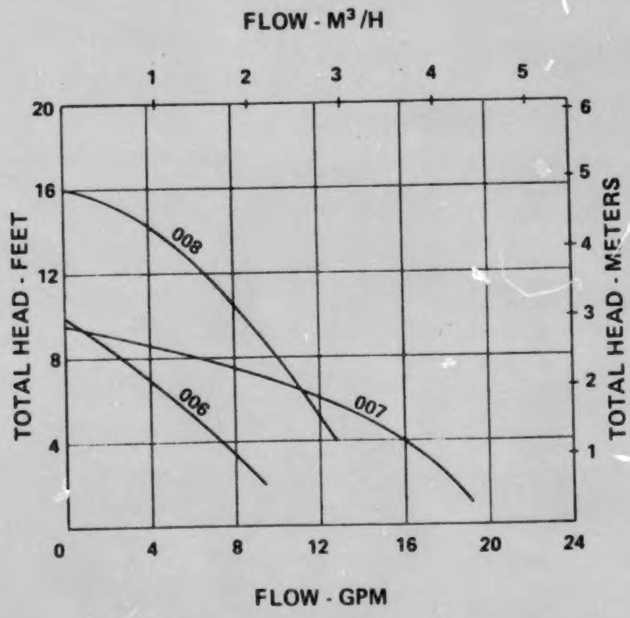
Options: None.

Installation Requirements: May be used with water or aqueous solutions of ethylene or propylene glycol at a maximum temperature of 240°F. Bronze version for open systems or drain down systems.

Maintenance Requirements: None.

Guarantee/Warranty: 1-yr. limited warranty.

Suggested List Price: $84. Consult manufacturer for trade and dealer discounts.



(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-6

Example IV: In examining the spec sheets for these two air handlers (Figures 10-7 and 10-8), we find that both will apparently deliver about the same amount of air. However, sometimes the decision is made in favor of available options or the fact that one may offer a wider range of information, i.e., BHP at various RPM and static pressure.

Solar Products Specifications Guide

SOLAR AIR HANDLER AH-15, AH-18

R-M Products
5010 Cook St.
Denver, CO 80216

Stephen Piro (303) 825-0203

TECHNICAL SPECIFICATIONS

Type: Internal dampers

Overall Dimensions: 60 in. x 29 in. x 29 in. (AH-15, AH-18)

Weight: 275 lbs

Does Unit Have Built-in Controller: Yes

Insulation: Yes R-3

Capacity Range: See chart on back

Domestic Hot Water Coil Available: Yes

Can Unit be Mounted in Positions Other Than Upright? Right, left and horizontal mount

Components not Supplied for Installation: Thermostat or sensor wires (hardware is included)

Limitations: Those limitations inherent in single fan systems

PRODUCT DESCRIPTION

A solar air handler designed for easy installation, maintenance and operation while operating through the four modes of solar air heating.

Features: Completely prewired (Honeywell controls); mode operational lights; external motor bearing shaft, etc., for easy maintenance and reduced thermal wear. Internal filters.

Options: 7 day timeclock (day-night) occupied and unoccupied thermostats. Domestic hot water coil.

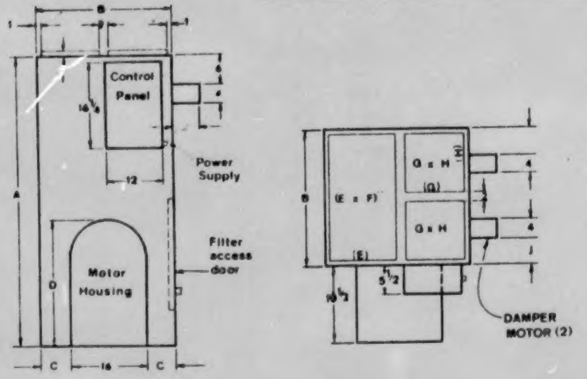
Installation Requirements: Unit must be set; three duct connections; run thermostat wires, sensor wires, power wiring and fused disconnect.

Maintenance Requirements: Change filters, high temp grease (every 3-4 months).

Guarantee/Warranty: 1 yr. parts and workmanship, existing factory warranties on component parts.

Suggested List Price: $1,200 - $2,200

MODEL NUMBER	In. dia.	A	B	C	D	E x F	G x H	J
Model 15AH	15 inches	60	29	6½	27	12x27½	12x12½	3½
Mode - 18AH	18 inches	60	29	6½	27	12x27½	12x12½	3½



Models 15AH and 18AH weigh approximately 275 lbs

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-7

SOLAR AIR HANDLER AH-15, AH-18

MODEL-15AH

	Outlet	1/8" S.P.		1/4" S.P.		3/8" S.P.		1/2" S.P.		5/8" S.P.		3/4" S.P.		1" S.P.		1-1/4" S.P.		1-1/2" S.P.	
CFM	Vel.	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
553	500							961	.17	1051	22	1136	.27	1283	.39	1413	.51	1536	.64
664	600					902	.15	1002	.20	1092	25	1175	.31	1321	.43	1447	.56	1568	.70
774	700			859	.14	955	.18	1047	.24	1130	29	1214	.35	1348	.48	1487	.62	1602	.77
885	800	842	.12	931	.17	1015	.22	1096	.28	1175	.34	1254	.40	1395	.54	1523	.69	1639	.84
995	900	919	.16	1006	.21	1083	.18	1156	.33	1229	.39	1303	.46	1437	.60	1568	.75	1679	.92
1106	1000	999	.21	1084	.27	1154	.33	1224	.39	1291	.45	1356	.52	1484	.67	1601	.83	1718	1.00
1217	1100	1080	26	1161	.33	1230	.39	1293	.46	1355	.53	1429	.60	1538	.76	1651	.92		
1327	1200	1158	.32	1242	40	1297	.47	1366	.54	1423	61	1482	.69	1593	.85	1700	1.02		
1438	1300	1238	.39	1320	.48	1386	56	1442	.63	1497	.71	1551	.79	1653	.96				
1548	1400	1314	.46	1402	.57	1464	.65	1521	.74	1572	.82	1622	.90						
1659	1500	1385	.53	1483	.67	1545	.77	1599	.85	1648	.94	1697	1.03						

MODEL-18AH

	Outlet	1/8" S.P.		1/4" S.P.		3/8" S.P.		1/2" S.P.		5/8" S.P.		3/4" S.P.		1" S.P.		1-1/4" S.P.		1-1/2" S.P.	
CFM	Vel.	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
995	900					749	.23	826	.30	900	.38	964	.46	1085	.65	1192	.84	1291	1.05
1106	1000					773	.26	847	33	920	.42	992	.51	1115	.69	1216	.90	1355	1.11
1217	1100			730	.22	806	29	876	.37	948	.46	1009	.55	1128	.75	1253	.96	1320	1.19
1327	1200			771	.26	839	.34	873	42	967	.51	1034	.60	1136	.81	1254	1.02	1351	1.26
1438	1300	741	23	813	.30	875	.38	[illegible]	.47	1000	.56	1061	.66	1173	.87	1161	1.12	1373	1.34
1548	1400	786	.27	855	.35	916	44	975	52	1031	.62	1084	.72	1199	.94	1299	1.17	1395	1.42
1659	1500	836	.32	897	.41	956	.49	1013	.59	1064	.69	1119	.79	1223	1.01	1322	1.25	1414	1.51
1770	1600	876	.37	944	.47	999	.56	1050	.66	1094	.76	1149	.87	1254	1.09	1347	1.34		
1880	1700	920	.43	988	.53	1041	.63	1093	.73	1140	.84	1190	.95	1283	1.19	1375	1.44		
1991	1800	964	49	1030	.61	1082	.71	1134	.82	1181	.93	1227	1.04	1319	1.28	1406	1.54		
2101	1900	1007	.55	1076	.69	1128	.80	1175	.91	1221	1.02	1264	1.14	1349	1.39				
2212	2000	1049	.63	1120	.77	1172	.90	1218	1.01	1261	1.13	1304	1.25	1387	1.50				

Figure 10-7, continued

SOLAR AIR MOVER SAM-10A, SAM-20A, SAM-30A

Solar Control Corp.
5721 Arapahoe
Boulder, CO 80303

Liz Quinn (303) 449-9180

TECHNICAL SPECIFICATIONS

Type: Internal dampers

Overall dimensions: 36 x 24 x 36 in

Weight: 180 - 200 lb

Does unit have built-in controller: Yes

Insulation: Yes, R-4

Capacity range: 500 to 2200 cfm at static pressures of 0.25 in to 1.5 in

Domestic hot water coil available: Yes

Can unit be mounted in positions other than upright? Must be mounted upright

Components not supplied for installation: Backdraft dampers, ductwork, collectors

Limitations: None

PRODUCT DESCRIPTION

Provides total system air flow and operational mode control in one compact package. Contains blower, motor and all powered dampers necessary to mechanize the system, as well as built-in differential controller. Fully compatible with conventional furnaces, thermostats, air conditioners, and heat pumps.

Features: Two-speed motor, domestic hot water preheat control.

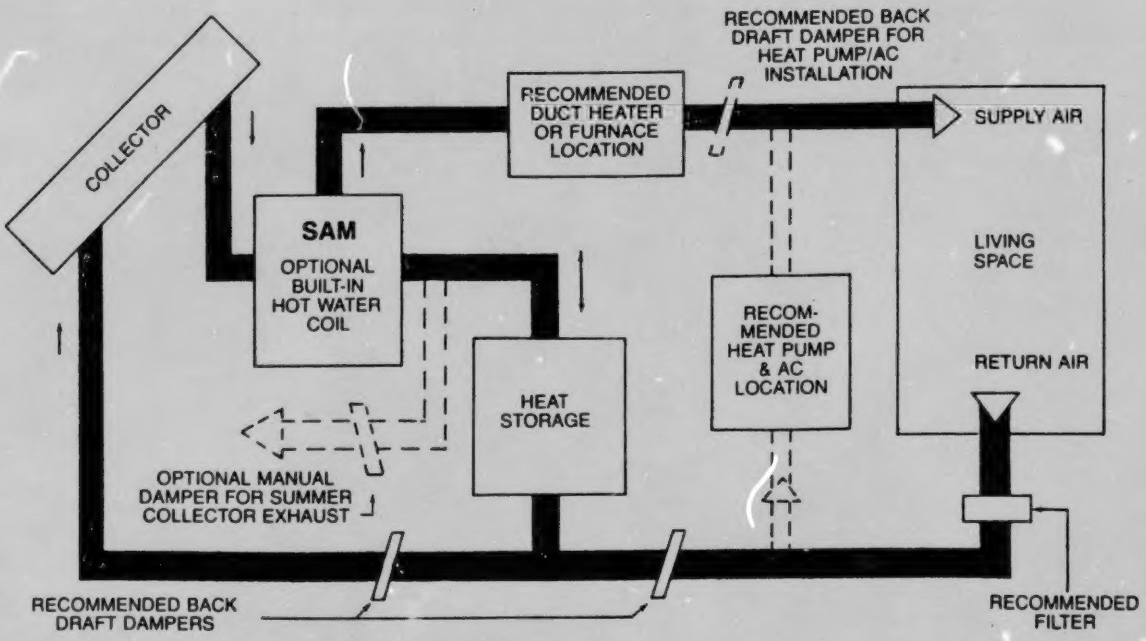
Options: Domestic water preheat coil.

Installation Requirements: Minimum 2 ft. clearance on 3 specific sides.

Maintenance Requirements: Visual inspection.

Guarantee/Warranty: 1-yr. warranty

Suggested List Price: SAM-10A, $1398.
SAM-20A, $1598.
SAM-30A, $1798.


RECOMMENDED BACK DRAFT DAMPER FOR HEAT PUMP/AC INSTALLATION
COLLECTOR
RECOMMENDED DUCT HEATER OR FURNACE LOCATION
SUPPLY AIR
SAM
OPTIONAL BUILT-IN HOT WATER COIL
LIVING SPACE
RECOM-MENDED HEAT PUMP & AC LOCATION
RETURN AIR
HEAT STORAGE
OPTIONAL MANUAL DAMPER FOR SUMMER COLLECTOR EXHAUST
RECOMMENDED BACK DRAFT DAMPERS
RECOMMENDED FILTER

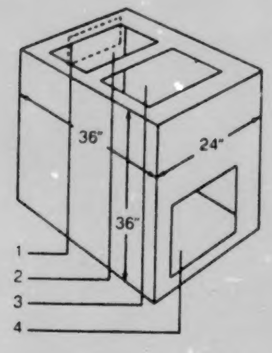
(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-8

SOLAR AIR MOVER SAM-10A, SAM-20A, SAM-30A

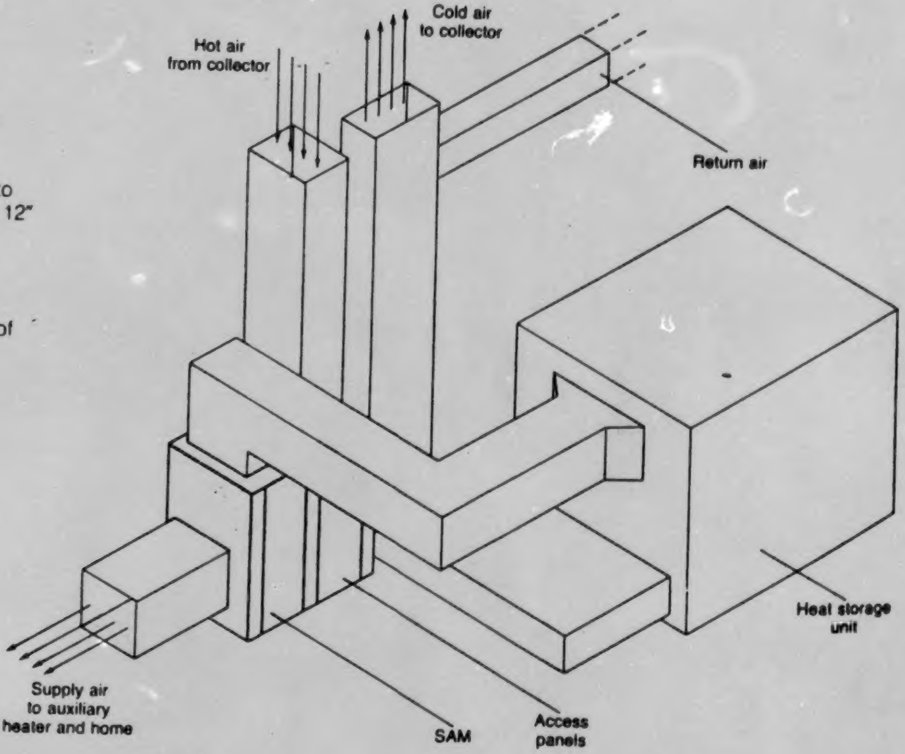
MODEL	BLOWER			STATIC PRESSURE						
	H/P	Volts	Amps	.25"	.5"	.75"	1.0"	1.25"	1.5"	2.0"
SAM-10	½	115	10 max	1600	1300	1200	1100	700	—	—
SAM-20	¾	115	14 max	2100	1800	1600	1400	1100	500	—
SAM-30	1½	115	16 max	—	3000	2200	2000	1800	1500	800

Underlined areas are preferred operating range.


36"
24"
36"
1
2
3
4

1. SAM control unit
2. Inlet for hot air duct from collector—19" x 12"
3. Inlet/outlet for duct to heat storage—19" x 12"
4. Outlet to auxiliary heater/house—19" x 15"

Two access doors are provided on each side of the Solar Air Mover.


Hot air from collector
Cold air to collector
Return air
Heat storage unit
Supply air to auxiliary heater and home
SAM
Access panels

(This information is provided by the manufacturer, who is responsible for technical accuracy.)

Figure 10-8, continued

As one can see, from examining the specification sheets, it is at the point - equipment selection - in the design process where one must use all of the information and skills previously attained in the foregoing courses and previous modules of this course. The selection criteria has been established (per application) and must be an integral part of the selection process.

One final word on making a decision from manufacturers' specification sheets: Examine them carefully. The choice may hinge upon some critical point that is not apparent to the untrained eye. Be aware of equipment, manufacturers' and material limitations. When in doubt, get another opinion.

V-S-305

SIZING, DESIGN AND RETROFIT

RETROFIT CONSIDERATIONS

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

RETROFIT CONSIDERATIONS

Retrofit Concepts

Applying solar heating and cooling to existing buildings can provide immediate reduction in our fossil and nuclear energy demands. Unfortunately, our inventory of tens of millions of buildings and related structures is of less interest to professional designers and energy administrators than new buildings are. For building owners, and particularly the homeowner whose heating and cooling bills are doubling, and then doubling again, the retrofitting of existing buildings should have top priority.

As with new buildings, retrofitting old ones can be done on varying levels of technological complexity, monetary and energy expense, and common sense. At one end of the scale of complexity and expense are the five schools which were solar powered through the initiative of the National Science Foundation. In the middle range is one of the first solar heated houses, located in Boulder, Colorado, and designed by Dr. George Löf in 1950.

The end of the scale presently applicable for most home retrofitting includes the simpler and often more efficient methods. There are three basic ways of retrofitting buildings (see Figure 9-1). One way is to apply the collectors to existing, or slightly modified, exterior surfaces of the building; that is, the walls or roofs. Another way is to attach them to an addition onto the building, such as a porch, garage, or a new wing. A third way is to build a structure separated from the building. This might be an auxiliary out-building, such as an unattached shed, garage, or barn, or a structure built for the sole purpose of supporting the collector. This method will be discussed later.

Because of the unusual constraints of existing buildings, the size orientation and tilt angle of the collector may be predetermined. Often the economic constraints of trying to alter existing conditions restrict the optimization of the design. For collectors which heat domestic hot water, the design is somewhat more flexible because the collectors are smaller and are used year round; the sun position varies much more during twelve months than during the shorter heating season. Collectors for cooling have difficulty attaining adequate efficiency under the best of conditions and should conform to optimum design as much as possible, making application to existing buildings difficult. For space heating, the size can be as small as 100 square feet or larger than half the floor area of the building. For domestic hot water heating, it can be as small as necessary or as large as 30 square feet per person.

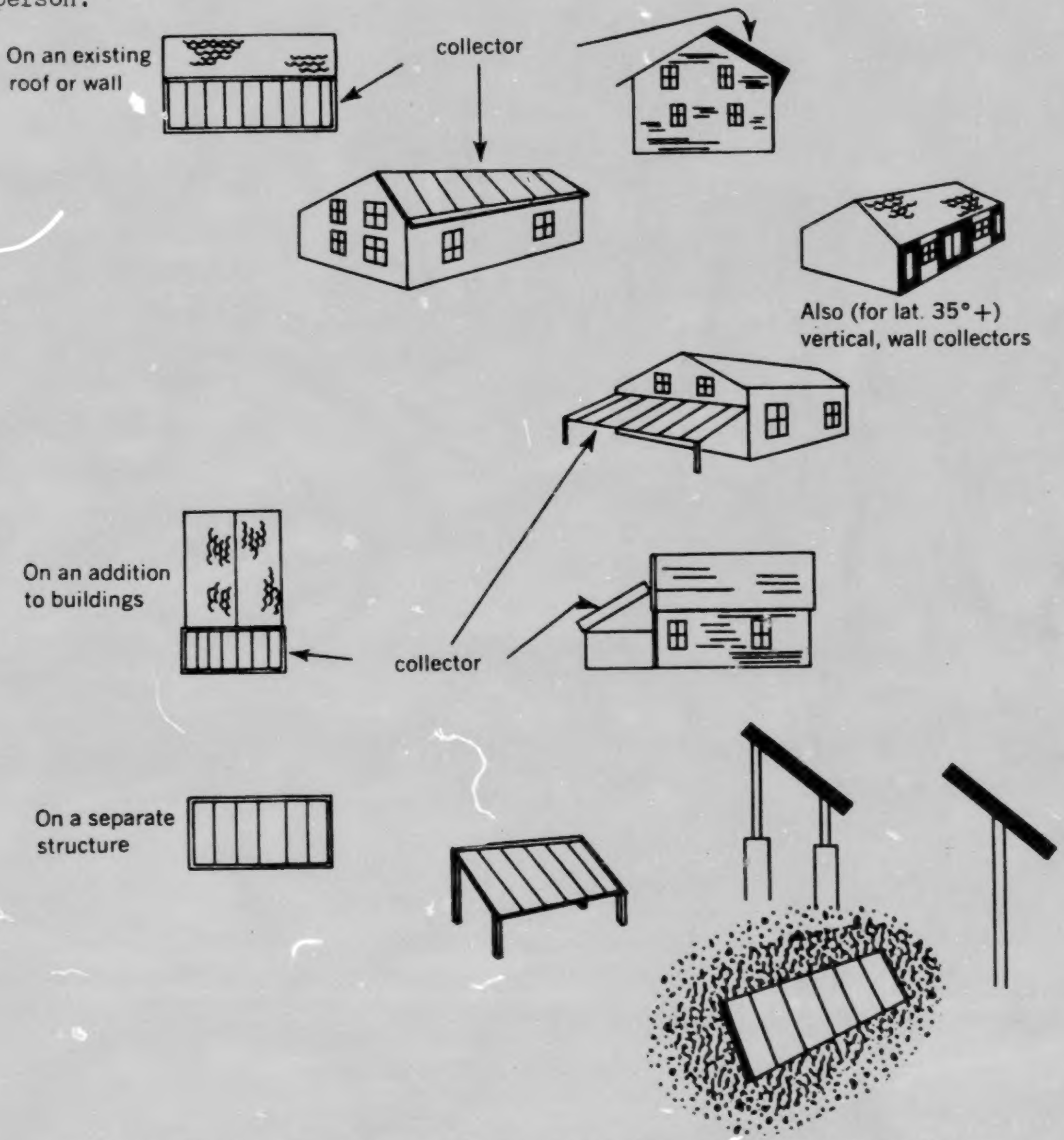

On an existing roof or wall
collector
Also (for lat. 35°+) vertical, wall collectors
On an addition to buildings
collector
On a separate structure

Figure 9-1: The Application of Solar Collectors to Existing Buildings.

The orientation of the collectors can range from south-southeast to south-southwest for space heating, and from southeast to southwest for domestic hot water heating. For space heating collectors, the tilt angle (measured from the horizontal) can vary from an angle of the latitude to an angle of latitude plus 55°. For 40° N, the range then is from 40° to 90° (vertical). For domestic hot water heating collectors, the tilt can range from latitude minus 10° to latitude plus 25°. For 40° N, this allows a range of from 30° to 75°.

In all the above ranges, the seasonal or yearly overall efficiency will not vary more than 10 or 20% from the optimum. One of the easiest ways of collecting solar heat with existing roofs is to pass water or air over the shingled surface. The surface should be as black as possible, painted if necessary, and free from debris. Frames for two layers of glass or the equivalent (such as fiberglass-reinforced polyester) are attached to the rafters, taking care to prevent leaks.

The roof could also be covered with corrugated aluminum painted black and covered by glass. Water is released through a perforated pipe along the ridge and collected in a gutter, or the equivalent. Dr. Thomason experimented with this method and found it to be relatively inefficient. However, if the roof is an existing one, the small cost involved in converting it to a solar collector might justify a low efficiency.

Figure 9-2 shows some possible design details. Portions of south-facing walls could be converted to air-type collectors in a fashion similar to that for a

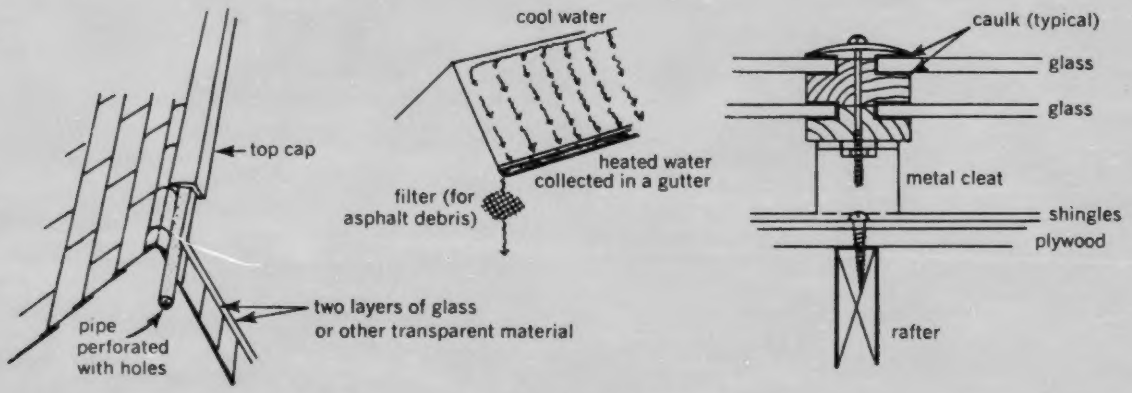

top cap
pipe perforated with holes
two layers of glass or other transparent material
cool water
heated water collected in a gutter
filter (for asphalt debris)
caulk (typical)
glass
glass
metal cleat
shingles
plywood
rafter

Figure 9-2: Converting an existing roof to an open flow water-type solar collector.

roof. Water-type collectors would be less practical in this case because of the absence of a sloping surface over which the water can trickle.

Separate structures can support collectors in yards, although the aesthetic appearance may not appeal to most homeowners. An example of such a device which can be easily set up and dismantled is shown in Figure 9-3. Cool air from the house is blown out through the bottom of the window to the solar collector and back through the top of the window. The installation is similar to a window air conditioner. Better control is obtained by ducting the cool air from one window and returning the warm air through another.

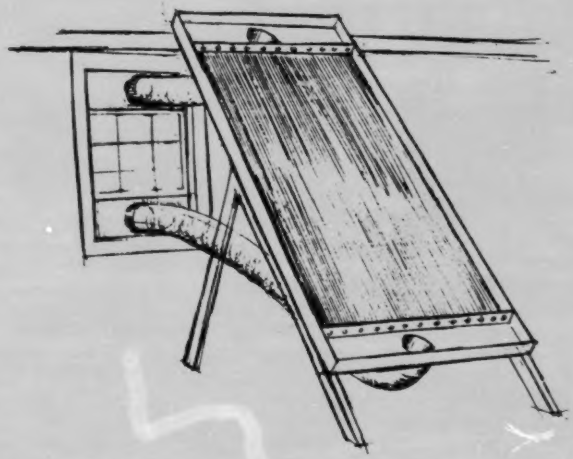


Figure 9-3: A portable, in the yard air-type solar collector.

A quick but modest effort can be made in retrofitting existing buildings by constructing simple windowbox air-type solar collectors. Figures 9-4, 9-5, and 9-6 are modifications of the vertical, thermosiphoning solar collector. They are designed to be incorporated into the openings of existing windows. Figure 9-4 is a design credited to Buck Rodgers of Embudo, New Mexico. The cool air from the room is drawn into the collector by the warmed air leaving it. The vertical variation of this, Figure 9-5, is particularly applicable to large buildings.

Althought the windowbox collector can be of almost any size, its effectiveness, although substantial on a per-square-foot basis, really is almost immeasurable unless it is significantly larger than the window. If a collector size of

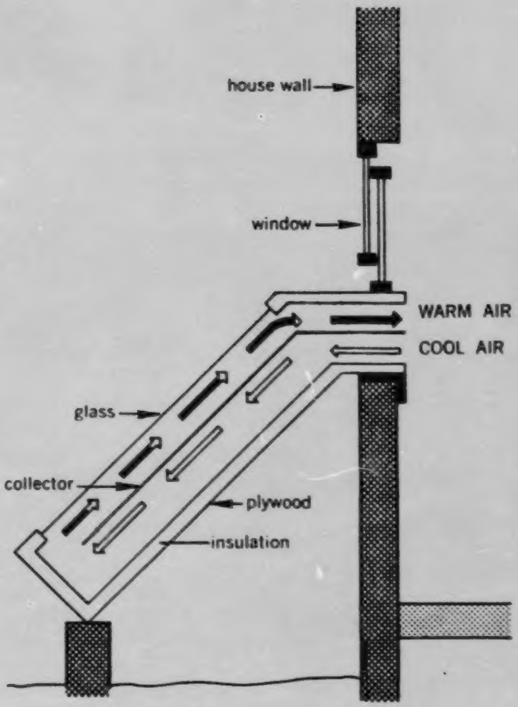

house wall
window
WARM AIR
COOL AIR
glass
collector
plywood
insulation

Figure 9-4: Window box solar collector, invented by Buck Rodgers, Embudo, New Mexico (Taken from Alternative Sources of Energy, #13)

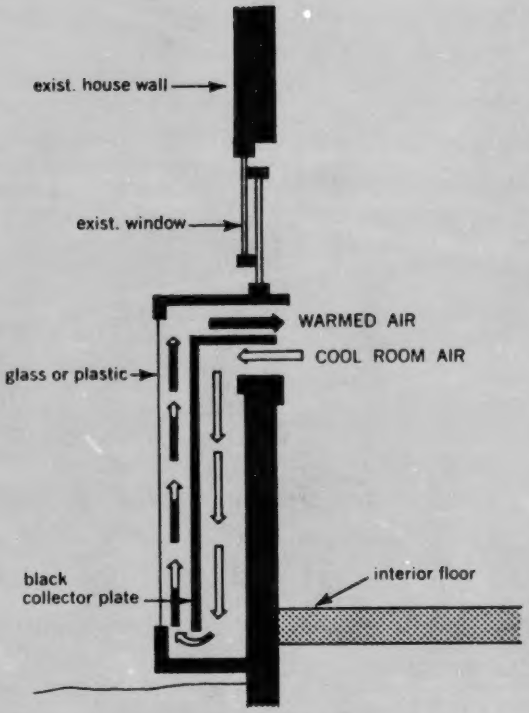

exist. house wall
exist. window
WARMED AIR
COOL ROOM AIR
glass or plastic
black collector plate
interior floor

Figure 9-5: A variation of the window box collector.

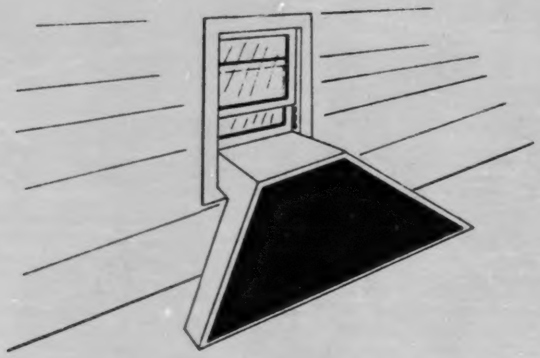


Figure 9-6: The window box: a frontal view.

one-quarter to one-half the building floor area is required to provide 50% of the heating, it should be clear that large collectors are required for a noticeable effect on overall energy savings. Figure 9-6 shows one method of making the collector larger than the window.

The difficult task of adding heat storage to existing buildings has been practically solved by J. P. Gupta and R. K. Chopra of the Defense Laboratory, Jodhpur, India. They developed a simple solar room heater which requires no mechanical power and which can be incorporated into existing buildings. As can be seen in Figure 9-7, the solar collector is resting against the building and faces south. A tall, uninsulated hot water tank stands inside the room with its back against, but well insulated from, the outside wall. Water circulates by thermosiphoning action from the flat plate collector, up to the tank, and back down to the collector. For climates with freezing weather, anti-freeze is added to the water. Heat radiates from the front of the tank to the room.

The range of low impact, low cost alternatives has only been skimmed here. Innovations and new designs are desperately needed, as is the development of mechanisms which will encourage quick adoption by every segment of the construction industry.

The foregoing examples of retrofitting existing buildings to solar applications are simple, low technology methods that are usually economically desirable.

The solar industry must be charged with the responsibility of developing systems that are more aesthetically appealing, if this vast inventory of existing structures is to be fully utilized.

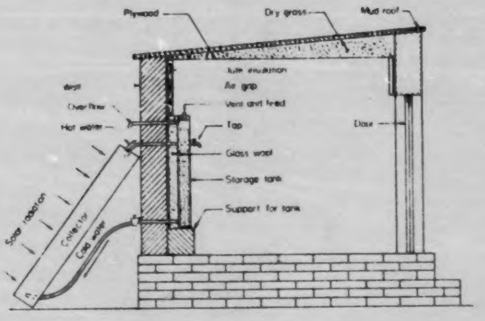

Plywood
Dry grass
Mud roof
Jute insulation
Air gap
Wall
Overflow
Vent and feed
Hot water
Tap
Door
Glass wool
Storage tank
Support for tank
Solar radiation
Collector
Cold water

Figure 9-7: A simple solar room heater. (Designed by J. P. Gupta and R. K. Chopra of the Defense Laboratory, Jodhpur, India).

GENERAL CONSIDERATIONS

Retrofit, in the present context, means the adaptation of solar systems to existing buildings. Solar system designs for existing buildings are fundamentally the same as for new buildings. However, there are installation factors that need to be considered in retrofitting; factors that are not involved in new construction. These factors relate to the structural and mechanical features of existing buildings and to the cost of installation. The most important thing to remember is that each installation is a special case, and generalizations of problems are difficult.

INSULATING EXISTING BUILDINGS

Although insulating a building is not strictly a feature of solar energy systems, it has a significant impact on solar systems with energy conservation designs in buildings. Many existing residential buildings have little or no insulation in the walls and ceilings. If a solar system is contemplated, an initial step is to insulate the building.

If the cost for adequately insulating the building is high, an economic analysis to determine benefits and cost is recommended.

TREES AND LANDSCAPE

The availability of sunshine for the particular building is of prime importance. There are many existing residential buildings that have been landscaped generously with trees for the specific purpose of shading the building. (See Figure 9-8 on the following page). The trees that block sunlight to the collector array will have to be removed. Although solar radiation will filter through leafless branches of deciduous trees during the winter, the reduction in useful sunshine could greatly affect the system size and performance. An alternative to removal is to reduce the height of the trees, but this will invite an annual or periodic maintenance cost that is chargeable to the solar system.

There are many locations where buildings on hillsides are shaded by neighboring structures. Solar systems for buildings that are shadowed part of the day will necessitate an unusual orientation of the collectors, with consequent increase in collector area and system cost.

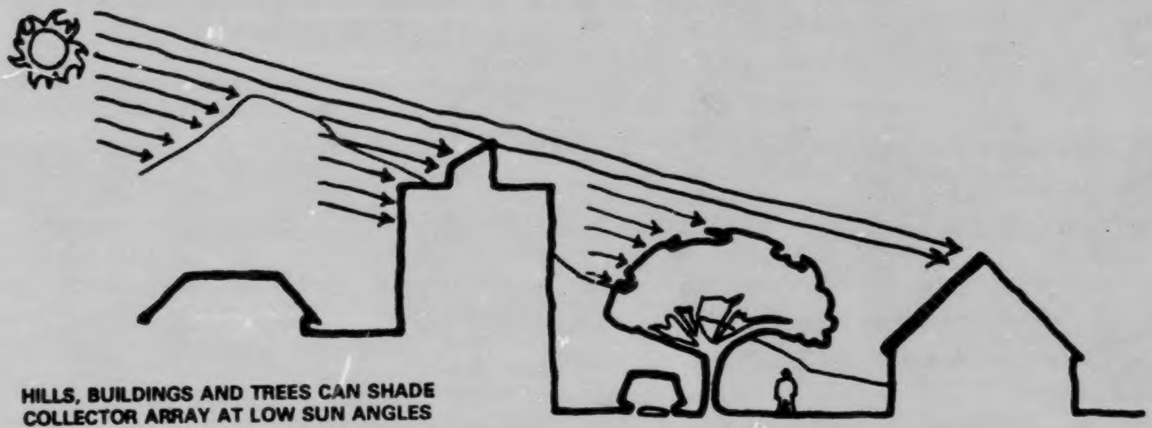

HILLS, BUILDINGS AND TREES CAN SHADE
COLLECTOR ARRAY AT LOW SUN ANGLES

Figure 9-8: Trees and Landscape

SPACE HEATING

There are several potential difficulties involved in providing retrofit solar space heating systems. These problems concern:

1. Collector location;
2. Equipment location;
3. Adaptation to the existing heating system.

COLLECTOR LOCATION

Collectors can be advantageously mounted on the roof of new buildings if the weight of the collectors can be supported. Otherwise, collectors will have to be supported on a separate structure on the ground. In new construction, the roof pitch is usually set at the desired collector tilt angle to maximize the collection of solar energy for a particular orientation. In retrofit situations, the roof pitch normally is 5 in 12, or 22.6° from horizontal. This angle is too flat for solar collectors in most locations, so a separate frame is needed to mount the collectors at a more suitable angle. One possible arrangement is illustrated in Figure 9-9, which will appear on the following page.

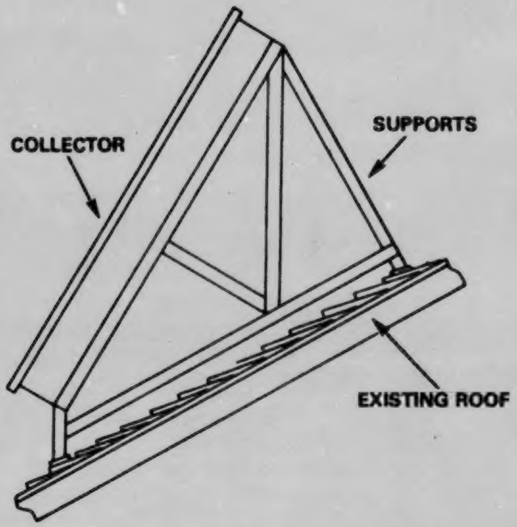


Figure 9-9: Collector Location

The "add-on" appearance of the collectors and supports may be aesthetically unsatisfactory to some homeowners. When aesthetics govern, either the entire roof must be reconstructed to blend them architecturally with the building, or the collectors must be placed at ground level. Removal and reconstruction are expensive. Although there may be beneficial effects in the renovation other than to accommodate collectors, the costs will be chargeable to the solar system.

When collectors cannot be placed on the building roof, they must be placed at ground level, preferably on the south side and adjacent to the building. Placing collectors at ground level offers some advantages and some disadvantages.

One advantage is lower pumping head for open loop systems. Another is that piping and ducting to the collector banks are easier to install than in the attic of an existing building with a low-pitched roof. Maintenance of collectors at ground level is easier.

A disadvantage is that the collector array may offer hazards to the occupants. Another disadvantage is pipes and ducts are in unheated areas, or exposed to the outside air. Insulation around pipes and ducts must be thick to reduce heat losses.

EQUIPMENT LOCATION

The location of equipment needed for solar heating systems may offer difficulties for some retrofit installations. The bulkiest equipment that has to be installed is the storage tank for a hydronic system, and a rock bed for an air system. The most easily accessible area in the building is at ground floor level. Ground floor space is expensive compared to equivalent space in the basement or garage.

The fabrication of storage tanks and rock boxes in basements, or placing of rocks in storage are restrictive activities in retrofit installations. The walls of rock bed storage containers can be fabricated relatively easily, but fabricating tanks for water storage could be more difficult. Tanks may be fabricated inside either by welding or bolting sections together. If bolted tanks are used, neoprene or butyl rubber lining is recommended to prevent leaks from the bolted seams.

Locating the storage tank or rock bed in the garage offers the simplest installation for retrofit situations. Adaptation to the existing heating system with the storage tank in the garage may require longer pipes and ducts than if storage were located inside the building. The biggest disadvantage with storage located in the garage is that the heat loss from storage is not recovered as useful heat in the building enclosure.

It is recommended that heat exchangers and pumps be located close to the storage tank for hydronic systems, to minimize heat losses and economize of space. Other equipment, such as pumps and heat exchangers, will not occupy much space. Maintenance will also be facilitated if all the equipment is located in one place.

ADAPTATION TO EXISTING HEATING EQUIPMENT

The solar heating systems discussed in this module are for central air distribution systems. Adaptation of solar systems to existing buildings is likewise facilitated if a central distribution system exists. While baseboard heating systems are prevalent in many non-solar hydronic systems, flat-plate collectors will not function well with such systems. Fan coil units are recommended for such retrofit installations.

Adapting to an existing central forced air system requires the installation of the hydronic heating coil in the airstream. This coil is usually installed in the return air side. However, with proper controls, various configurations may be used.

A two-blower setup is most suitable for air solar system retrofit installations. The existing blower will have to operate independently of the heating element control, and the dampers will control the different modes of operation.

DOMESTIC HOT WATER

Addition of a solar potable water system to an existing home is far less complex than retrofitting for space heating. Since the demand is year-round, the tilt angle for the collectors is less critical as the sun's altitude angle changes drastically over a year; therefore, the existing roof angle can usually be tolerated.

The storage system is treated in the same manner as in the previous module on Service Water.

SPECIFIC CONSIDERATIONS

The installation of solar energy equipment to heat and/or cool an existing residence is a complex and expensive operation.

A careful analysis should be made to determine whether or not it is feasible, whether or not it is economically sensible, and whether or not it is, in fact, practicable.

First of all, the geographical location is important in determing what kind of space system; a space heating system; a space cooling system; or a combination space heating/cooling system.

In certain areas of the country, a solar space heating system is adequate for comfortable living. Even though some days in the summer are uncomfortably hot, they can be ignored in consideration of the expense of installation and operation of an air-conditioning system for such a short time. In other areas, a space cooling system is adequate for comfortable living. The relatively few cool days in the winter can be taken care of by auxiliary heating equipment.

For the majority of geographical areas, of course, a combination heating and cooling solar system, which in turn supplies most of the domestic hot water supply, can be a great aid in cutting down fuel costs. Once it has been decided what kind of heating, cooling, or heating/cooling system is wanted, it is then necessary to analyze the residence and property to see if retrofitting is feasible.

The Solar Collector Subsystem

The physical design, construction, and material properties of your roof are of primary importance to the positioning of solar collectors.

1. The pitch of the roof is quite probably several degrees off the optimum for your geographical position, which is approximately the latitude of your area. Situating the collector on a custom-built ramp or platform can solve this problem.

2. The structural rigidity of the roof may present a problem. Will the roof hold the solar collectors? What kind of collectors are needed? How much do they weigh?

3. The position of the roof in relation to the sun is a factor of great importance. The part of the roof on which the solar collectors are to be installed must have at least six hours of clear sunlight each day. Any less than that will produce inefficient results. Think of trees, hilly slopes, buildings, and so on.

4. The siting of the collectors must be analyzed to be sure they do not present hazards to power wirds, telephone wires, or neighbors. For example, if the reflection from the collectors shines directly into someone's window in the morning or afternoon, you must get assurances that there will be no hard feelings or lawsuits.

Optional Siting

If the roof proves unacceptable as a mount for solar collectors, you can always improvise other sitings.

For example, you can mount them on an existent fence or wall that faces to the south.

If you have a garage detached from the house and not shadowed by it, you can mount the collectors on its roof slope. One can even mount collectors on a south wall, keeping them vertical with some loss in efficiency, or letting them slope at the proper inclination by spreading out into the yard.

The Solar Storage Subsystem

Another important consideration in retrofitting is the installation of the storage tank or storage pit. In many cases, the addition of a storage tank for a fluid-medium collector system presents no problem. The addition of an extra tank in the basement or cellar can be achieved without total dislocation, and with a minimum of pipe installation.

However, it is possible that the house does not have a cellar or basement. If the existing hot water system and furnace is in a crowded space, which cannot be expanded without excessive structural changes, there is always the possibility that you can use part of your garage. If that is not possible, one may be able to bury the storage subsystem in the ground near the house.

Air Medium

A collector system of the air-medium type presents more of a problem in the storage subsystem than in the collector subsystem. The problem is obvious. Most air-medium systems store heat in pits of crushed rock. The large space needed for these rock pits is usually to be found under a house in the basement area, or in the garage.

If your cellar is already crowded, you may be able to locate the rock pit outside by burying it in the yard. Ducts connected to the rock pit will carry in hot air and pipe out warmed air on demand. This method is the least desirable.

No matter which kind of collector system you install, you can use the heating and/or cooling system as a backup for the new solar energy system.

The Solar Distribution System

The most efficient and most easily adaptable system of heating and cooling in the home is the forced-air system. This system delivers warmed air to every room of a house through a ductwork in walls and floors, moving colder air back to be expelled, filtered, and/or reheated.

Because the typical solar space heating and space cooling system recovers stored heat either directly through air pumped over heated rocks or through air passing over heated coils of hot water, the forced-air ductwork can usually be connected just as it is to a newly-installed solar collector and storage system. However, if the existing forced-air system was installed initially only for heating and not for heating and air conditioning, the size of the ducts may not be big enough to carry a solar heating supply.

If the forced-air system was installed for heating and cooling, the ductwork is adequate for a solar energy system and can immediately be connected to the new installation. In the event that the ductwork is too small, you will have to consider having it replaced with larger ducts. A heating and cooling engineer can give you advice on this.

If the existing system is a hot water baseboard radiation system, the solar energy system cannot be used without increasing the existing radiation area by about three to four times. Structurally, this is a rather unattractive prospect. Also, the design of the rooms will be changed considerably.

There is one solution, however, which may be a bit costly, but which does solve the problem. By adding small fan coils in the baseboard units, in order to boost their output, a baseboard system can be adapted to solar.

If you plan to install a solar system in conjunction with a heat pump system, you need only hook up the collector subsystem with the heat pump for an excellent and effective augmentation of the heat pump's performance.

PREPARING THE HOUSE FOR RETROFITTING

With considerations of solar collection area, storage area, and distribution area out of the way, you must check the house for its tightness and ability to prevent energy loss. If there are multiple air leaks, the addition of solar collectors will be a waste of time. The efficiency of solar energy can be reduced in half by careless weatherstripping, air leaks, and insufficient insulation.

To prepare any house for solar retrofitting, it is mandatory that the house be as airtight as possible, with proper "breathing" holes to prevent condensation. Making a house energy-fit is not easy; however, in these days of high fuel costs, even a house that is not using solar energy should be carefully and effectively sealed against heat leaks.

SUMMARY CONSIDERATIONS

Solarizing new construction can be done following established guidelines and accepted practices. Solar heating/cooling can be incorporated in the design. In retrofitting an existing structure, the following considerations may determine whether or not retrofitting is practical.

1. Collector placement - roof (which one?), or ground mounted; will roof support the added loads?
2. Siting - what potential infringements on solar radiation exist?
3. Orientation - may have serious effect on efficiency.
4. Storage type and location - will existing structure need to be "beefed-up" to handle the added load? Can it be placed in the desired location?
5. Fluid Transport System - installation problems.
6. Accessibility for service and maintenance.
7. Auxiliary Equipment - can the solar system be interfaced with existing equipment?
8. Control system - accessible and reliable. Don't make it too complex.

Any modification to existing structure or equipment in retrofitting must be added to the cost of solarizing the home.

V-S-323

SIZING, DESIGN AND RETROFIT

PROGRAMMED SYSTEM SIZING - ANALYSIS AND DESIGN

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

PROGRAMMED SYSTEM SIZING - ANALYSIS AND DESIGN

The use of programmable calculators and computers certainly decreases the time required to make a detailed analysis of the structure. The versatility inherent within a good program with input variables allows rapid decisions as to the effect of changing structural or system components.

It would not be practical because space would not allow a full treatment of the many programs available as design and analysis tools. Therefore, we will look at portions of two programs designed for the TI-59 programmable calculator.

As we examine the program and input variables, you will be able to gain a working knowledge of these programs. This knowledge can be applied to other instruments and programs by changing the commands where required.

THERMAL ANALYSIS PROGRAM

The thermal analysis probram is used to calculate a residence design heat loss and is based on the $UA \Delta T$ conduction model. The information presented here was taken directly from the ASHRAE Handbook of Fundamentals. The total heat loss is the sum of the transmission losses and the infiltration losses. The transmission losses consist of heat transmission through surfaces where a temperature gradient exists (e.g., walls, windows, ceilings, floors). The transmission heat loss, Q_{TRANS}, through a surface is given by:

$$Q_{TRANS} = AU(T_i - T_{WDT});$$

where

A is the area of the surface ft^2,

T_i is the enclosed air temperature, °F,

T_{WDT} is the winter design temperature °F, and

U is the overall heat-transmission coefficient = $1/\ R$ Btu/ (hr. - ft^2 - $^\circ$F)

where R_T is the total resistance of the separating layers. From a study of typical homes, built-in values for the resistance of typical roofs, waals, and ceilings were included in the program. These values are as follows:

$$\Sigma R_{wall} = 3.6$$

$$\Sigma R_{roof} = 4.0$$

$$\Sigma R_{ceiling} = 1.8$$

$$\Sigma R_{floor} = 2.6$$

These values do not include additional levels of insulation that can be installed. The user has the capability of adding insulation to the walls and ceiling. For example, if an additional 3-1/2 inches of insulation haring R-9 is put in the house wall, the ΣR_{wall} becomes

$$R_{wall} = 3.6 + 9 = 12.6$$

and

$$U_{wall} = 1/12.6 = 0.68$$

An unheated attic temperature is calculated from the following equation:

$$T_{attic} = \frac{(U_{ceiling} A_{ceiling} T_i) + (U_{roof} A_{roof} T_{WDT})}{(U_{ceiling} A_{ceiling}) + (U_{roof} A_{roof})}$$

In the program, T_i has been set at 70° F.

The total heat transmission loss is given by:

$$\begin{aligned} Q_{TRANS} = & \left[(U_{wall}\ A_{wall}) + (U_{window}\ A_{window})\right](T_i - T_{WDT}) \\ & + \left[(U_{ceiling}\ A_{ceiling})\ (T_i - T_{attic})\right] \\ & + \left[(U_{floor}\ A_{ceiling})(T_i - 55)\right] \end{aligned}$$

The third term in the previous equation is used to calculate the losses through the floor into an unheated basement or crawlspace.

Infiltration losses arise from the necessity to warm outside air, which enters the building or leaks in through cracks around windows, doors, basement and ceiling openings, etc. The heat, Q_{INFIL}, required to warm the infiltrating air from T_{WDT} to T_i is given by:

$$Q_{INFIL} = V \rho C_p (T_i - T_{WDT});$$

where

V is the hourly volume of external air entering the structure.

C_p is the specific heat of air and has a value of 0.24 Btu/lb-°F, and

ρ is the air density and is 0.075 lb/ft^3.

The above equation can be simplified to

$$Q_{INFIL} = 0.144\, V(T_i - T_{WDT})$$

In the program, the heat losses due to infiltration are calculated based upon an entered number of air changes per hour and the volume is eight times the floor area. Or:

$$Q_{INFIL} = 0.144\, (A_{floor})(70 - T_{WDT})(\text{No. of air changes/hour}).$$

The total heat loss is

$$Q_{TOTAL} = Q_{TRANS} + Q_{INFIL}\ (\text{Btu/hr}).$$

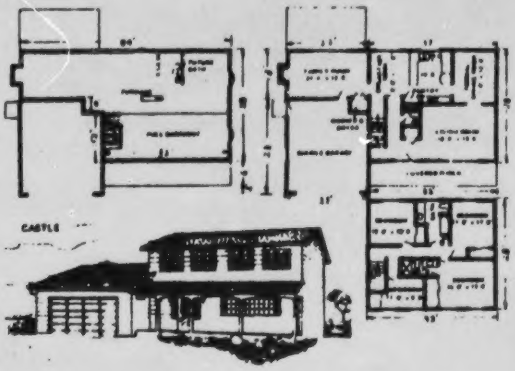
The equation to convert this heat loss to Btu per degree-day is given by:

$$Q_{DES} = \frac{Q_{TOTAL} \times 24}{65 - T_{WDT}}$$

No corrections are made for internal heat generation. This model should be adequate for quick calculations of the design heat loss. If more detailed calculations are required, detailed computer programs or other calculation procedures should be used.

THERMAL ANALYSIS EXAMPLE

To illustrate how this program can be used to calculate the design heat loss, consider the following example (see figure below).



Because the house is a complex shape, the problem will be solved in two stages. The first analysis will be of the two-story portion of the house, followed by an analysis of the one-story portion (the family room). The entire heat loss from the house is the sum of the losses from both portions.

For the two story portion, the floor plan is 32' x 28' with a window area of 10% of the floor area. The roof has a 5-12 pitch.

Gross wall area: $(32' \times 16' \times 2) + (28' \times 16')$
$+ (8' \times 28') - (5.83' \times 14' \times 1/2 \times 2)$
$= 1{,}614\ \text{ft}^2$.

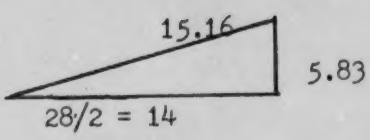
(At this time, assume no heat transfer into the garage from the house. This loss will be accounted for later).

Ceiling area: $A_{\text{ceiling}} = (32' \times 28') = 896\ \text{ft}^2$.

Window area: $A_{\text{window}} = (.1) \times (1{,}614') = 161\ \text{ft}^2$

Floor area: $A_{\text{floor}} = (32' \times 28' \times 2) = 1{,}792\ \text{ft}^2$

Roof area: (5-12 pitch) including the gable siding which encloses the attic.



$$A_{roof} = (15.16' \times 32' \times 2) + 5.83' \times 14' \times 1/2 \times 4$$
$$= 1{,}134 \text{ ft}^2$$

The house has single pane windows, i.e.,

$$U_{window} = 1.13$$

Now assume that the house is to be built with insulation levels of R-9 in the walls and R-19 in the ceilings:

$$R_{wall} = 9$$

$$R_{ceiling} = 19$$

and the house is to be located where the winter design temperature is -9° F.

$$T_{WDT} = -9^{\circ} \text{ F.}$$

To calculate the heat loss for the two-story portion of the building in Btu/Degree-Day, the following procedure is followed:

ENTER		PRESS	DISPLAY
1	Thermal Analysis Program Side 1		1.
4	Thermal Analysis Program Side 4		4.
1453	A_{wall}	$\boxed{\text{2nd}}$ $\boxed{\text{A'}}$	1453.
896	$A_{ceiling}$	$\boxed{\text{2nd}}$ $\boxed{\text{B'}}$	896.
161	A_{window}	$\boxed{\text{2nd}}$ $\boxed{\text{C'}}$	161.
1792	A_{floor}	$\boxed{\text{2nd}}$ $\boxed{\text{D'}}$	1792.
1134	A_{roof}	$\boxed{\text{2nd}}$ $\boxed{\text{E'}}$	1134.
-9	T_{WDT}	$\boxed{\text{A}}$	-9.
9	R_{wall}	$\boxed{\text{B}}$	9.
19	$R_{ceiling}$	$\boxed{\text{C}}$	19.
1.13	U_{window}	$\boxed{\text{D}}$	1.13
	Calculate Heat Loss	$\boxed{\text{E}}$	15189.

The heat loss is 15,189 Btu/DD from this portion of the house. To find the design heat loss rate, which is the heat loss in Btu/hr. when the outdoor temperature is the winter design temperature, the following keystrokes are entered.

ENTER	PRESS	DISPLAY
	RCL	
	1	
	1	46833.

and the design heat loss is 46,833 Btu/hr. for the two-story portion of the house.

For the one-story portion of the house, i.e., the family room:

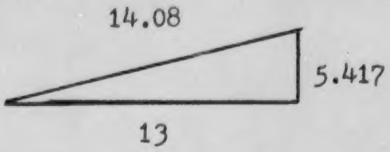
Gross wall area = $(22' \times 8') + (13' \times 8') = 280\ \text{ft}^2$

Wall area less windows: $A_{wall} = (.9) \times (280) = 252\ \text{ft}^2$

Ceiling area: $A_{ceiling} = (22' \times 13') = 286\ \text{ft}^2$

Window area: $A_{window} = (0.1) \times (280) = 28\ \text{ft}^2$

Floor area: $A_{floor} = (22' \times 13') = 286\ \text{ft}^2$



$$A_{roof} = (14.08' \times 22') = 310\ \text{ft}^2$$

ENTER		PRESS	DISPLAY
252	A_{wall}	2nd A'	252.
286	$A_{ceiling}$	2nd B'	286.
28	A_{window}	2nd C'	28.
286	A_{floor}	2nd D'	286.
310	A_{roof}	2nd E'	310.

(It is not necessary to repeat entry of T_{WDT}, R_{wall}, $R_{ceiling}$, or U_{window} as they are unchanged from the previous part of the problem.)

$\boxed{\text{E}}$ 2942

and, therefore, the total heat loss from this portion of the house is 2,942 Btu/DD. The design heat loss at the design temperature is found by

$\boxed{\text{RCL}}$

$\boxed{1}$

$\boxed{1}$ 9072.

which yields a design heat loss of 9,072 Btu/hr.

The total heat loss for both portions of the house is then

$$15{,}189 + 2{,}942 = 18{,}131 \text{ Btu/DD}$$

and the total design heat loss rate at the design temperature is

$$46{,}833 + 9{,}072 = 55{,}905 \text{ Btu/hr.}$$

This represents all of the building losses except for losses to the garage. These can be estimated by assuming a garage temperature when the outside temperature is the design temperature of -9° F. Choose a garage temperature of:

$$\frac{70 + (-9)}{2} = 30^{\circ} \text{ F}$$

and solve for the heat loss

$$Q = A_{wall} \times U_{wall} \times (70-30)$$
$$= [(22' + 14') \times 8'] \times \left(\frac{1}{12.6}\right) \times (40)$$
$$Q = 914 \text{ Btu/hr.}$$

and the design heat loss from the entire house at the design temperature is, therefore

$$55{,}905 + 914 = 56{,}819 \text{ Btu/hr.}$$

The total heat loss per degree day can be found by the formula

$$\frac{(\text{Btu/hr.}) \times 24}{65 - T_{WDT}} = \text{Btu/DD}$$

which becomes

$$\frac{56{,}819 \times 24}{65 - (-9)} = 18{,}428 \text{ Btu/DD}$$

Interestingly, the heat loss of the residence was calculated using an elaborate computer model and found to be 57,340 Btu/hr.

The default values for the number of air changes per hour and the inside temperature are 0.75 and 70° F, respectively. These can be altered after entering the program by the following keystrokes:

ENTER	PRESS	DISPLAY
For air changes/hour enter the number	$\boxed{\text{STO}}\boxed{0}\ \boxed{0}$	Value
For inside temperature enter the number	$\boxed{\text{STO}}\boxed{0}\ \boxed{1}$	Value

You may change the default values in the program by following the above steps and then rewriting the program card (side 4 only). This is accomplished by:

	$\boxed{\text{INV}}\ \boxed{\text{2nd}}\ \boxed{\text{fix}}$	
4 (Program Card Side 4)	$\boxed{\text{2nd}}\ \boxed{\text{write}}$	4.

SOLAR ANALYSIS EXAMPLE

The application of the FCHART (liquid) program will be illustrated by considering the design of a single family residential structure in Columbus, Ohio (latitude, 40.0° N). Assume that the design heat load is determined to be 63,000 Btu's/hour at a design ambient temperature of 2° F. This results in a

$$63{,}000\ \frac{\text{Btu's}}{\text{hour}} \times \frac{\text{24 hours}}{\text{day}} \times \frac{1}{(65-2)^{\circ}} = 24{,}000\ \frac{\text{Btu}}{\text{DD}}$$

24,000 Btu/degree-day house. Suppose that the potable hot water load is estimated to be 80 gallons/day (a good estimate is 20 gallons/day/occupant) to be raised from 52° F to 140° F, and this does not change throughout the year. Then the hot water load is

$$L_{PHW} = 80\ \frac{\text{gallons}}{\text{day}} \times 8.25\ \frac{\text{lbs}}{\text{gallon}} \times \frac{\text{1 Btu}}{\text{lb-F}^{\circ}} \times (140-52)^{\circ}\text{F} \times 30\ \frac{\text{days}}{\text{month}}$$

$$L_{PHW} = 1{,}742{,}400 \text{ Btu's/month.}$$

Flat plate collectors are to be used, and the design parameters are

$$F_R U_L = 0.83$$

and

$$F_R(\overline{\tau\alpha}) = 0.69$$

The collectors are to be mounted facing due south at a slope of 45°.

Table I, on the next page, lists the meteorological variables for Columbus, Ohio, and these values must be entered on a Meteorological Data Card. To do this, Program Write Data Card is used. Note: the calculator must be in degree mode for the program to operate properly.

```
       JAN     FEB     MAR     APR     MAY     JUN     JUL     AUG     SEP     OCT     NOV     DEC
                               NEW YORK        NY      LAT = 40.77
H      479.    734.   1069.   1360.   1593.   1733.   1692.   1434.   1220.    892.    542.    424.
TA     32.0    33.8    41.0    51.8    62.6    71.6    77.0    75.2    68.0    59.0    48.2    35.6
D/H    .502    .450    .415    .413    .405    .392    .390    .414    .397    .406    .483    .511
DD     973.    879.    750.    414.    124.      6.      0.      0.     27.    223.    528.    887.
                               ROCHESTER       NY      LAT = 43.12
H      498.    741.   1113.   1467.   1895.   2109.   2116.   1807.   1368.    929.    531.    420.
TA     24.8    24.8    33.8    46.4    57.2    66.2    71.6    69.8    62.6    51.8    41.0    28.4
D/H    .436    .412    .378    .372    .330    .310    .295    .311    .335    .362    .447    .459
DD    1271.   1126.    992.    567.    285.     46.      9.     26.    126.    398.    735.   1138.
                               SCHENECTADY     NY      LAT = 42.83
H      479.    741.   1010.   1253.   1526.   1655.   1637.   1471.   1106.    807.    476.    383.
TA     23.0    24.8    33.8    46.4    59.0    68.0    73.4    69.8    62.6    51.8    39.2    28.4
D/H    .463    .420    .424    .441    .421    .411    .403    .398    .427    .427    .506    .512
DD    1339.   1154.    977.    543.    244.     36.      8.     20.    137.    422.    756.   1181.
                               SYRACUSE        NY      LAT = 43.12
H      476.    712.   1069.   1379.   1784.   2035.   2057.   1736.   1320.    889.    457.    380.
TA     23.0    24.8    33.8    46.4    57.2    66.2    71.6    69.8    62.6    51.8    41.0    28.4
D/H    .455    .429    .395    .398    .355    .325    .308    .328    .350    .381    .513    .504
DD    1283.   1131.    986.    555.    272.     46.     11.     18.    120.    392.    720.   1144.
                               CLEVELAND       OH      LAT = 41.40
H      457.    664.   1150.   1390.   1928.   2061.   2031.   1818.   1386.    970.    520.    424.
TA     28.4    30.2    35.6    48.2    59.0    68.0    71.6    69.8    64.4    53.6    41.0    32.0
D/H    .509    .484    .379    .401    .325    .320    .314    .313    .340    .364    .488    .495
DD    1181.   1039.    896.    501.    244.     40.      9.     17.     95.    354.    702.   1076.
                               COLUMBUS        OH      LAT = 40.00
H      476.    730.   1091.   1449.   1799.   2072.   1998.   1759.   1556.   1054.    656.    487.
TA     30.2    32.0    39.2    51.8    60.8    69.8    73.4    71.6    64.4    53.6    41.0    32.0
D/H    .514    .457    .409    .388    .355    .317    .321    .330    .298    .342    .408    .459
DD    1088.    949.    809.    426.    171.     27.      0.      6.     84.    347.    714.   1039.
                               DAYTON          OH      LAT = 39.90
H      597.    826.   1224.   1537.   1910.   2120.   2079.   1895.   1563.   1139.    689.    535.
TA     28.4    30.2    39.2    51.8    60.8    71.6    75.2    73.4    66.2    55.4    41.0    30.2
D/H    .423    .410    .365    .365    .331    .307    .304    .299    .298    .314    .394    .428
DD    1144.    969.    806.    413.    166.     13.      0.      7.     63.    307.    696.   1057.
                               PUT-IN-BAY      OH      LAT = 41.65
H      442.    734.   1077.   1360.   1821.   1998.   2090.   1902.   1471.   1088.    579.    409.
TA     28.4    30.2    35.6    48.2    59.0    69.8    75.2    73.4    66.2    57.2    42.8    32.0
D/H    .521    .438    .405    .410    .349    .333    .301    .293    .314    .314    .439    .508
DD    1190.   1036.    902.    510.    219.     24.      0.      0.     40.    277.    660.   1060.
                               OKLAHOMA CITY   OK      LAT = 35.40
H      940.   1169.   1501.   1836.   1991.   2297.   2249.   2168.   1784.   1397.   1047.    874.
TA     35.6    39.2    48.2    59.0    68.0    77.0    80.6    80.6    73.4    60.8    48.2    39.2
D/H    .298    .311    .307    .303    .315    .269    .268    .244    .260    .263    .277    .294
DD     874.    664.    532.    180.     36.      0.      0.      0.     12.    148.    474.    775.
                               STILLWATER      OK      LAT = 36.15
H      763.   1054.   1431.   1601.   1851.   2186.   2186.   1998.   1678.   1298.    947.    752.
TA     35.6    41.0    48.2    60.8    68.0    77.0    80.6    80.6    71.6    62.6    48.2    39.2
D/H    .373    .348    .322    .338    .346    .292    .282    .282    .285    .290    .310    .344
DD     865.    644.    517.    174.     38.      0.      0.      0.     10.    146.    465.    772.
                               TULSA           OK      LAT = 36.20
H      763.   1007.   1360.   1611.   1899.   2149.   2101.   1958.   1615.   1213.    863.    719.
TA     37.4    41.0    48.2    60.8    68.0    77.0    82.4    80.6    73.4    62.6    50.0    39.2
D/H    .373    .364    .343    .356    .335    .300    .300    .291    .302    .319    .349    .363
DD     800.    666.    528.    176.     28.      0.      0.      0.     10.    143.    468.    781.
                               ASTORIA         OR      LAT = 46.20
H      339.    575.    984.   1368.   1807.   1777.   1972.   1681.   1324.    778.    420.    288.
TA     41.0    42.8    42.8    46.4    51.8    55.4    59.0    59.0    57.2    51.8    46.4    41.0
D/H    .539    .478    .405    .389    .346    .381    .324    .333    .329    .400    .490    .562
DD     786.    599.    639.    516.    394.    255.    163.    151.    201.    378.    555.    688.
```

ENTER		PRESS	DISPLAY
1	Write Data Card (Side 1)		1.
	Initialize Program	E	0.
40	Latitude	D	40.
.514	$\bar{D}/\bar{H}$	A	0.514
476	$\bar{H}$	B	476.
1088	DD(degree-days)	C	1088.

That is, for January, these are the pertinent data entered. For February, the following keystrokes are entered:

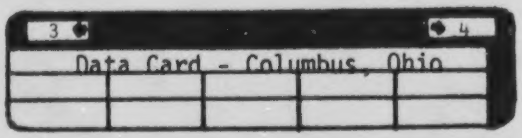
.457	$\bar{D}/\bar{H}$	A	0.457
730	$\bar{H}$	B	730.
949	DD	C	949.

This procedure is followed for each month for the entire year. To write the Meteorological Data Card, enter the following keystroke. See if you can produce a data card for Columbus, Ohio.

ENTER	PRESS		DISPLAY
	2nd A'	(and enter a blank magnetic card into the card reader.)	3.

ENTER	PRESS	DISPLAY
	2nd B' (turn the card and enter the card used above)	4.

This procedure writes the data entered onto a blank magnetic card. The card should be labeled Data Card - Columbus, Ohio, with 3 and 4 shown in corners as shown below.



To check that the card has been correctly written, the latitude should be displayed by RCL 0 0 . The twelve values of $\overline{D}/\overline{H}$, $\overline{H}$, and DD are stored contiguously in memory locations 1 through 12, 13 through 24, and 25 through 36 respectively.

The remaining calculation is to determine the fraction of the heating load provided by the solar heating system. Let us address this question by considering five different collector areas: 500 ft^2, 750 ft^2, 1,000 ft^2, 1,250 ft^2, and 1,500 ft^2.

ENTER		PRESS	DISPLAY
1	Solar Analysis (Liquid-Side 1)		1.
2	Solar Analysis (Liquid-Side 2)		2.
3	Data Card-Columbus (Side 3)		3.
4	Data Card-Columbus (Side 4)		4.

ENTER		PRESS	DISPLAY
45	Collector Tilt Angle	$\boxed{\text{2nd}}$ $\boxed{A'}$	45.
0.83	F_RU_L	$\boxed{\text{2nd}}$ $\boxed{B'}$	0.83
0.69	$F_R(\overline{\tau\alpha})$	$\boxed{\text{2nd}}$ $\boxed{C'}$	0.69
24000	Q_{DES}	$\boxed{\text{2nd}}$ $\boxed{D'}$	24000.
1742400	Hot Water Load	$\boxed{\text{2nd}}$ $\boxed{E'}$	1742400.
500	Collector Area	$\boxed{A}$	500.
	Initialize	$\boxed{B}$	0.
	Calculate Load Fraction F	$\boxed{E}$	0.13

The fraction of the load provided by a system with 500 ft^2 of collectors for the month of January is 0.13. For February

F	$\boxed{E}$	0.24

Continuing in this fashion for each month, the Solar Fraction of the Load Worksheet can be filled in as shown. To obtain the fraction for the year, the following keystrokes are entered:

F(YEAR)	$\boxed{C}$	0.37

and the total heating load is

Total Load	$\boxed{D}$	156748800.

To calculate the fraction provided by a different collector area, the entire procedure is repeated with the different collector area input and re-initialization of the program as shown below.

ENTER	PRESS	DISPLAY
New Collector Area	$\boxed{A}$	
Initialization	$\boxed{B}$	0.
Calculate F	$\boxed{E}$	

See if you can duplicate the worksheet results (following page).

Note that a different hot water load can be entered each month by entering the hot water load before calculating the monthly load provided by the solar system.

WORKSHEET - SOLAR FRACTION OF LOAD

COLLECTOR PARAMETERS:

$F_R U_L$ = 0.83

$F_R(\overline{\tau\alpha})$ = 0.69

STRUCTURE:

Q_{DES} = 24,000 BTU/DD

L_{PHW} = 1,742,000 BTU'S/MONTH

Columbus, Ohio
45° Tilted Colle
tor
Liquid System

V-S-340

FRACTION OF LOAD PROVIDED BY SOLAR HEATING SYSTEM

MONTH	AREA= 500 FT^2	AREA= 750 FT^2	AREA= 1000 FT^2	AREA= 1250 FT^2	AREA= 1500 FT^2
JAN	0.13	0.19	0.25	0.30	0.35
FEB	0.24	0.33	0.42	0.50	0.56
MAR	0.35	0.49	0.60	0.69	0.76
APR	0.62	0.79	0.89	0.95	1.00
MAY	0.95	1.00	1.00	1.00	1.00
JUN	1.00	1.00	1.00	1.00	1.00
JUL	1.00	1.00	1.00	1.00	1.00
AUG	1.00	1.00	1.00	1.00	1.00
SEP	1.00	1.00	1.00	1.00	1.00
OCT	0.72	0.88	0.96	1.00	1.00
NOV	0.32	0.45	0.55	0.64	0.70
DEC	0.18	0.25	0.32	0.38	0.44
YEAR	0.37	0.47	0.54	0.60	0.65

Q = 156,748,000 Btu/year

POTABLE HOT WATER SYSTEM EXAMPLE

The FCHART procedure may also be used to design potable hot water systems when no solar space heating is desired. Such a case will be illustrated using the same residence in Columbus, Ohio (Latitude 40° N.) that was used in the "Solar Analysis Example." However, in this example, only potable hot water will be supplied by solar energy. The energy required each month to meet the hot water load is given as:

$$L_{PHW}\text{ (Btu's/month)} = (G)\,\frac{\text{gallons}}{\text{day}} \times (8.3)\,\frac{\text{lbs.}}{\text{gallon}} \times (1)\,\frac{\text{Btu}}{\text{lb.-}^{\circ}\text{F}}$$

$$\times\ (T_{set} - T_M)\ ^{\circ}\text{F} \times (30)\,\frac{\text{days}}{\text{month}}$$

where G is the hot water usage rate in gallons/day; T_{set} is the set temperature of the hot water desired in ° F; and T_M is the temperature of the cold water entering the system from the water main in °F. If the hot water usage rate is not known, it may be estimated as 20 gallons/day/occupant for an average residence. For an average family of four, then, the hot water use rate would be 80 gallons/day. Most residential systems supply potable hot water at 140° F and an average value for water temperature from the main is 52° F.

For the Columbus family of four, then, the average hot water load each month is

$$L_{PHW} = (80) \times (8.3) \times (1) \times (140-52) \times (30)$$

$$L_{PHW} = 1{,}752{,}960\text{ Btu's/month}$$

assuming T_{set} and T_{main} as 140° F and 52° F, respectively.

The hot water load may also be changed for each month of the year by entering different values of T_M and the number of gallons of water used per day.

With the monthly potable hot water load known, the FCHART program may be used to calculate the fraction provided by solar. Let us address this question by considering five different collector areas: 30 ft^2, 45 ft^2, 60 ft^2, 75 ft^2, and 90 ft^2.

ENTER		PRESS	DISPLAY
1	Solar Analysis (Potable Hot Water Side 1)		1.
2	Solar Analysis (Potable Hot Water Side 2)		2.
3	Data Card-Columbus (Side 3)		3.
4	Data Card-Columbus (Side 4)		4.
45	Collector Tilt Angle	$\boxed{\text{2nd}}$ $\boxed{\text{A'}}$	45.
0.83	$F_R U_L$	$\boxed{\text{2nd}}$ $\boxed{\text{B'}}$	0.83
0.69	$F_R(\overline{\tau\alpha})$	$\boxed{\text{2nd}}$ $\boxed{\text{C'}}$	0.69
52	T_{MAIN}	$\boxed{\text{2nd}}$ $\boxed{\text{D'}}$	52.
80	Daily Hot Water Consumption	$\boxed{\text{2nd}}$ $\boxed{\text{E'}}$	80.
30	Collector Area	$\boxed{\text{A}}$	30.
	Initialize	$\boxed{\text{B}}$	0.
	Calculate Load Fraction F	$\boxed{\text{E}}$	0.10

The fraction of the load provided by a system with 30 ft^2 of collectors for the month of January is 0.10. For February

	F	$\boxed{\text{E}}$	0.18

Continuing in this fashion for each month, the Solar Fraction of the Load Worksheet can be filled in as shown. To obtain the fraction for the year, the following keystrokes are entered.

ENTER	PRESS	DISPLAY
F(YEAR)	$\boxed{C}$	0.27

and the total heating load is

Total Load	$\boxed{D}$	21120000.

To calculate the fraction provided by a different collector area, the entire procedure is repeated with the different collector area input and re-initialization of the program as shown below.

New Collector Area	$\boxed{A}$	
Initialization	$\boxed{B}$	0.
Calculate F	$\boxed{E}$	

See if you can duplicate the worksheet.

Note that a different hot water load can be entered each month by entering the main temperature and/or the daily water consumption before calculating the monthly load provided by the solar system.

WORKSHEET - SOLAR FRACTION OF LOAD

COLLECTOR PARAMETERS:

$F_R U_L$ = 0.83

$F_R (\overline{\tau\alpha})$ = 0.69

STRUCTURE:

Q_{DES} = 0 BTU/DD

L_{PHW} = BTU'S/MONTH

Columbus, Ohio
45° Collector Til
Liquid System
80 Gallons/Day
T_M = 52°F

FRACTION OF LOAD PROVIDED BY SOLAR HEATING SYSTEM

MONTH	F AREA= 30 FT²	F AREA= 45 FT²	F AREA= 60 FT²	F AREA= 75 FT²	F AREA= 90 FT²
JAN	0.10	0.14	0.19	0.23	0.27
FEB	0.18	0.25	0.32	0.38	0.44
MAR	0.24	0.35	0.43	0.51	0.58
APR	0.29	0.41	0.51	0.60	0.68
MAY	0.33	0.46	0.57	0.66	0.74
JUN	0.35	0.49	0.61	0.71	0.79
JUL	0.35	0.49	0.61	0.71	0.79
AUG	0.36	0.50	0.62	0.71	0.80
SEP	0.39	0.54	0.66	0.76	0.84
OCT	0.31	0.43	0.54	0.63	0.71
NOV	0.20	0.29	0.36	0.43	0.49
DEC	0.13	0.19	0.25	0.30	0.35
YEAR	0.27	0.38	0.47	0.55	0.62

Q = 21,120,000 Btu/year

APPLICATIONS OF THE PROGRAMS TO A SYSTEM SIZING PROBLEM

In this section, we will illustrate the application of the previously-described programs to a solar system sizing problem.

Suppose that a solar system is to be designed for a house to be located in the Washington, D. C., area (See Table I on the following page). We wish to consider both air and water systems. We would like to determine the economically optimum collector area required for a solar system for space heating and service hot water requirements.

The parameters to be used are as follows. An air system having

$$F_R\overline{\tau\alpha} = 0.52$$

$$F_R U_L = 0.52 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

is to be used. They are for the Solaron collectors. The collectors are to be tilted at 45 degrees and faced due south.

The service hot water load is to be considered as 80 gallons per day to be raised from 60° F to 140° F.

The house is to have R-19 for the walls, R-30 for the ceilings, and R-2 for the windows. No internal heat gains are to be considered. The house has 2,240 ft^2 of wall area, 1200 ft^2 of ceiling area, 160 ft^2 of window area, 1200 ft^2 of floor area, and 1697 ft^2 of roof area.

Economic parameters to be considered are:

Mortgage period = 25 years

Interest rate = 9%

Initial system cost = $13.50 per square foot of installed collector, plus $3,500 fixed costs.

Conventional fuel cost = $12.00 per million Btu.

```
 #     JAN     FEB     MAR     APR     MAY     JUN     JUL     AUG     SEP     OCT     NOV     DEC
                             DENVER              CO     LAT = 39.75
H      940.   1246.   1607.   1913.   2146.   2411.   2334.   2183.   1821.   1364.    966.    804.
TA     30.2    32.0    37.4    48.2    57.2    66.2    73.4    71.6    62.6    51.8    39.2    32.0
D/H    .221    .228    .244    .271    .280    .246    .250    .230    .223    .228    .248    .253
DD    1088.    902.    868.    525.    253.     80.      0.      0.    120.    408.    768.   1004.
                             GRAND JUNCTION      CO     LAT = 39.12
H      855.   1198.   1585.   1965.   2230.   2610.   2474.   2142.   1847.   1394.    970.    796.
TA     26.6    32.0    41.0    51.8    60.8    71.6    77.0    75.2    66.2    53.6    39.2    30.2
D/H    .278    .254    .254    .259    .261    .201    .218    .241    .217    .221    .253    .266
DD    1209.    907.    729.    387.    146.     21.      0.      0.     30.    313.    786.   1111.
                             GRAND LAKE          CO     LAT = 40.27
H      782.   1154.   1560.   1888.   2035.   2330.   2212.   1862.   1755.   1331.    863.    678.
TA     15.8    19.4    24.8    32.0    42.8    50.0    55.4    53.6    46.4    37.4    26.6    17.6
D/H    .299    .258    .253    .275    .303    .264    .276    .305    .238    .232    .290    .317
DD    1556.   1322.   1296.    945.    685.    450.    276.    313.    504.    803.   1176.   1476.
                             PUEBLO              CO     LAT = 38.28
H     1003.   1298.   1644.   2006.   2219.   2470.   2389.   2208.   1858.   1453.   1084.    881.
TA     30.2    33.8    39.2    51.8    60.8    69.8    77.0    75.2    66.2    53.6    41.0    33.8
D/H    .212    .224    .243    .252    .265    .233    .238    .227    .220    .208    .208    .233
DD    1082.    848.    775.    405.    148.     28.      0.      0.     55.    335.    726.    992.
                             HARTFORD            CT     LAT = 41.93
H      583.    833.   1209.   1419.   1748.   1969.   1950.   1692.   1342.    973.    608.    697.
TA     24.8    26.6    35.6    48.2    59.0    68.0    73.4    69.8    62.6    51.8    41.0    28.4
D/H    .398    .382    .356    .391    .366    .339    .331    .342    .352    .359    .415    .276
DD    1246.   1070.    911.    519.    226.     24.      0.     12.    106.    384.    711.   1141.
                             WASHINGTON          DC     LAT = 38.85
{H     586.    848.   1180.   1486.   1648.   2057.   1950.   1703.   1353.   1036.    778.    542.}
{TA    35.6    37.4    44.6    55.4    64.4    73.4    77.0    75.2    69.8    59.0    48.2    37.4}
{D/H   .448    .411    .387    .382    .392    .320    .332    .346    .365    .363    .359    .441}
{DD    871.    762.    626.    288.     74.      0.      0.      0.     33.    217.    519.    834.}
                             APALACHICOLA        FL     LAT = 29.75
H     1080.   1342.   1626.   2028.   2245.   2179.   1995.   1869.   1696.   1541.   1228.    973.
TA     53.6    55.4    59.0    66.2    73.4    78.8    80.6    80.6    78.8    69.8    60.8    55.4
D/H    .314    .304    .304    .271    .261    .288    .319    .321    .309    .267    .278    .330
DD     347.    260.    180.     33.      0.      0.      0.      0.      0.     16.    153.    319.
                             GAINESVILLE         FL     LAT = 29.65
H     1025.   1353.   1641.   1907.   2160.   2006.   1917.   1873.   1637.   1357.   1172.    936.
TA     55.4    57.2    62.6    69.8    75.2    78.8    80.6    80.6    78.8    71.6    62.6    57.2
D/H    .336    .301    .300    .281    .279    .324    .336    .320    .324    .325    .299    .347
DD     295.    240.    132.     19.      0.      0.      0.      0.      0.     13.    124.    258.
                             JACKSONVILLE        FL     LAT = 30.42
H      984.   1276.   1560.   1895.   2050.   1936.   1925.   1755.   1412.   1220.   1010.    848.
TA     53.6    55.4    60.8    68.0    73.4    78.8    80.6    80.6    77.0    69.8    60.8    53.6
D/H    .343    .319    .318    .301    .303    .340    .335    .346    .383    .363    .353    .377
DD     348.    282.    176.     24.      0.      0.      0.      0.      0.     19.    161.    317.
                             KEY WEST            FL     LAT = 24.55
H     1206.   1512.   1807.   2109.   2135.   2002.   1969.   1847.   1641.   1453.   1224.   1077.
TA     69.8    69.8    73.4    77.0    80.6    82.4    84.2    84.2    82.4    78.8    73.4    69.8
D/H    .321    .292    .280    .261    .282    .317    .320    .329    .339    .328    .331    .347
DD      16.     25.      5.      0.      0.      0.      0.      0.      0.      0.      0.     18.
                             MIAMI               FL     LAT = 25.78
H     1265.   1534.   1810.   2006.   2035.   1958.   1980.   1873.   1648.   1434.   1305.   1176.
TA     66.2    66.2    69.8    73.4    77.0    80.6    80.6    82.4    80.6    77.0    71.6    68.0
D/H    .289    .276    .274    .283    .304    .329    .319    .322    .334    .327    .292    .297
DD      74.     56.     19.      0.      0.      0.      0.      0.      0.      0.      0.     65.
                             PENSACOLA           FL     LAT = 30.47
H      922.   1183.   1493.   1877.   2072.   2094.   1980.   1877.   1585.   1453.   1025.    826.
TA     51.8    53.6    59.0    68.0    73.4    78.8    80.6    80.6    77.0    69.8    59.0    53.6
D/H    .370    .350    .336    .305    .299    .306    .323    .318    .335    .288    .347    .388
DD     427.    323.    211.     37.      0.      0.      0.      0.      0.     32.    189.    359.
```

rate of increase
of conventional = 7% per year.
fuel cost

The procedure to be followed in conducting the sizing study is illustrated in Figure 11-1, on the following page. Each step on this flowchart is illustrated in the following material.

A. Thermal Analysis

The first step in the process is to determine the design heat load for the building. This may be accomplished by using the Thermal Analysis Program presented earlier and illustrated below.

STEP	PROCEDURE	ENTER	PRESS		DISPLAY
1	Enter Program (Side 1)	1			1.
2	Enter Program (Side 4)	4			4.
3	Input Wall Area	2240	2nd	A'	2240.
4	Input Ceiling Area	1200	2nd	B'	1200.
5	Input Window Area	160	2nd	C'	160.
6	Input Floor Area	1200	2nd	D'	1200.
7	Input Roof Area	1697	2nd	E'	1697.
8	Input Winter Design Temperature	19	A		19.
9	Input Wall Insulation	19	B		19.
10	Input Ceiling Insulation	30	C		30.
11	Input Window U Factor	0.5	D		0.5
12	Calculate Q_{DES}		E		12705.

By following the above steps, we obtain

$$Q_{DES} = 12705 \text{ Btu/DD}$$

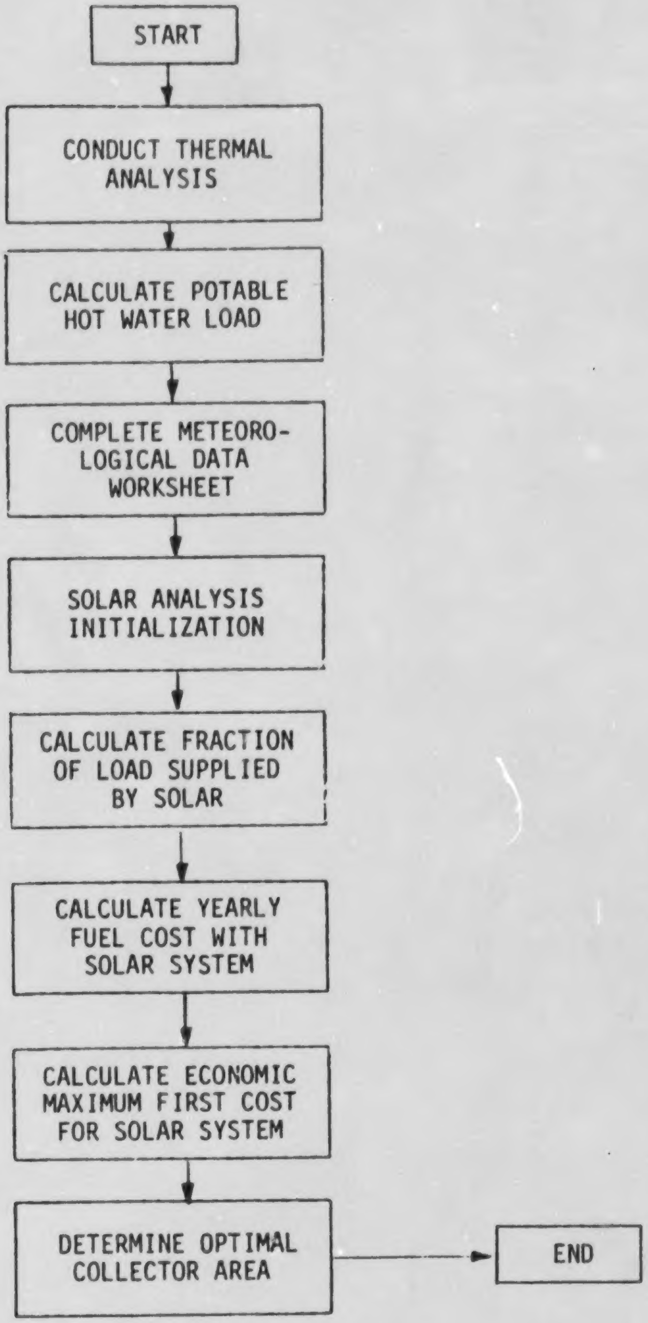

START
CONDUCT THERMAL ANALYSIS
CALCULATE POTABLE HOT WATER LOAD
COMPLETE METEOROLOGICAL DATA WORKSHEET
SOLAR ANALYSIS INITIALIZATION
CALCULATE FRACTION OF LOAD SUPPLIED BY SOLAR
CALCULATE YEARLY FUEL COST WITH SOLAR SYSTEM
CALCULATE ECONOMIC MAXIMUM FIRST COST FOR SOLAR SYSTEM
DETERMINE OPTIMAL COLLECTOR AREA
END

Figure 11-1: Flowchart of Sizing Procedure

The design heat loss in Btu/hr. may be determined by the following keystrokes.

RCL	
1	
1	24352.

The resulting figure is seen to be 24352 Btu/hr. at the winter design temperature of 19° F.

B. Service Hot Water Load

The next step in the design procedure is to calculate the service hot water requirements. A reasonable assumption for a single-family residence is 20 gallons of hot water per person per day. Assuming four people, we obtain for the potable hot water load:

$$L_{PHW} = (80 \frac{gal}{day})(8.33 \frac{lb}{gal})(1 \frac{Btu}{lb\text{-}^{\circ}F})(140^{\circ}\ F - 60^{\circ}\ F)$$

$$= 53312 \text{ Btu/day}$$

The average monthly requirement is, therefore

$$L_{PHW} = (53{,}312 \text{ Btu/day}) \times (30 \text{ days/month})$$

$$L_{PHW} = 1{,}599{,}360 \text{ Btu/month.}$$

This heating requirement must be added to the monthly space heating requirements.

C. Writing the Meteorological Data Card

The next step is to complete the meteorological data card for use by the FCHART program. The values for $\overline{D}/\overline{H}$, $\overline{H}$, DD and site latitude are obtained from Table I.

The process of writing the data card is described on the following page.

STEP	ENTER	PRESS	DISPLAY
1	Write Data Card (Side 1)		1.
2	Initialize	$\boxed{E}$	0.
3	38.51 (Latitude)	$\boxed{D}$	38.51
4	0.448 ($\overline{D}/\overline{H}$)	$\boxed{A}$	0.448
5	586 ($\overline{H}$)	$\boxed{B}$	586.0
6	871 (DD)	$\boxed{C}$	871.0

The corresponding $\overline{D}/\overline{H}$, $\overline{H}$, and DD figures for the remaining months are entered by repeating steps 4 through 6 for each month. After entering the December data, a data card can be written by pressing

$\boxed{2nd}$ $\boxed{A'}$	(Enter data card)	3.
$\boxed{2nd}$ $\boxed{B'}$	(Enter turned data card)	4.

D. Fraction of Load Supplied by Solar

The next step is to calculate the fraction of the total heating load that will be supplied by the solar system for various collector areas. Collector areas to be considered are 100, 200, 400, 600 and 800 ft^2.

First, we must load the solar analysis program. This is described as follows. We will demonstrate it for the air system.

STEP	ENTER	PRESS	DISPLAY
1	1 Solar Analysis (Air-Side 1)		1.
2	2 Solar Analysis (Air-Side 2)		2.

STEP	ENTER	PRESS	DISPLAY
3	3 Data Card-Washington, D.C. (Side 3)		3.
4	4 Data Card-Washington, D. C. (Side 4)		4.
5	45. (Tilt)	2nd A'	45.
6	0.52 (F_RU_L)	2nd B'	0.52
7	0.52 ($F_R(\overline{\tau\alpha})$)	2nd C'	0.52
8	12705 (Q_{DES})	2nd D'	12705.
9	1599360 (L_{PHW})	2nd E'	1599360.
10	600 (Area)	A	600.
11	Initialize	B	0.
12	Calculate F	E	0.45

We see that the solar system with 600 ft^2 of collector will provide 45% of the January heating requirements. The remaining monthly fractions may be determined by repeating step 12 for each month. After completing these calculations for the entire 12 month period, the fraction of the load provided by the solar system for the entire 12 month period can be obtained by entering the following keystroke.

PRESS	DISPLAY
C	0.71

To obtain the yearly total load, enter the following

D	72858240.

All results are shown on the following table.

WORKSHEET - SOLAR FRACTION OF LOAD

COLLECTOR PARAMETERS:

$F_R U_L$ = 0.52 Btu/Hr-Ft^2-°F

$F_R(\overline{\tau\alpha})$ = 0.52

STRUCTURE:

Q_{DES} = 12,705 BTU/DD

L_{PHW} = 2,599,360 BTU'S/MONTH

Washington, D.C.
45° Tilted Collector
Air System

V-S-352

FRACTION OF LOAD PROVIDED BY SOLAR HEATING SYSTEM

MONTH	AREA= 100 FT^2	AREA= 200 FT^2	AREA= 400 FT^2	AREA= 600 FT^2	AREA= 800 FT^2
JAN	0.08	0.16	0.31	0.45	0.57
FEB	0.12	0.23	0.44	0.62	0.77
MAR	0.16	0.32	0.58	0.80	0.95
APR	0.31	0.57	0.94	1.00	1.00
MAY	0.56	0.93	1.00	1.00	1.00
JUN	0.90	1.00	1.00	1.00	1.00
JUL	0.89	1.00	1.00	1.00	1.00
AUG	0.88	1.00	1.00	1.00	1.00
SEP	0.72	1.00	1.00	1.00	1.00
OCT	0.35	0.64	1.00	1.00	1.00
NOV	0.18	0.35	0.63	0.85	0.99
DEC	0.08	0.17	0.32	0.45	0.58
YEAR	0.23	0.38	0.58	0.71	0.80

Q = 72,858,240 Btu/year

E. Yearly Fuel Costs

The next step in the analysis procedure is to calculate the yearly fuel costs. We found in the previous step that the total annual heating requirement is 72,858,240 Btu's. Assume that the cost of electricity is $12.00 per million Btu's and that the conventional heating system is electric resistance heating. Therefore, the annual cost of heating without the solar system would be $874.30. The annual heating cost, if solar were installed, would be (1-F) times $874.30, where F is the fraction of the load supplied by solar as determined on the Solar Fraction of Load worksheet. The annual savings would be given by F * 874.30. For the system with 600 ft^2 of collector (F=0.71), the fuel cost for the solar heating system is $253.55, and the annual savings is $620.75.

F. Economic Maximum First Cost (EMFC) Calculation.

This is the final step in the analysis process. The maximum first cost that one should pay for a solar system to meet the particular requirements is determined in this step. The results for the 600 ft^2 collector are illustrated below.

STEP	PROCEDURE	ENTER	PRESS	DISPLAY
1	Enter EMFC (Side 1)	1		1.
2	Enter EMFC (Side 2)	2		2.
3	Enter EMFC (Side 3)	3		3.
4	Enter EMFC (Side 4)	4		4.
5	Input Mortgage Interest Rate	.09	$\boxed{\text{STO}}$ $\boxed{0}$ $\boxed{3}$	0.09
6	Input Mortgage Period	25	$\boxed{\text{STO}}$ $\boxed{0}$ $\boxed{2}$	25.
7	Input Collector Area	600	$\boxed{\text{A}}$	600.
8	Input Load	72.85824	$\boxed{\text{E}}$	72.85824
9	Input F	0.71	$\boxed{\text{C}}$	0.71
10	Input Fuel Cost	12	$\boxed{\text{D}}$	12.

ENTER	PROCEDURE	ENTER	PRESS	DISPLAY
11	Residence		2nd A'	0.
12	Calculate Economic Maximum First Cost		E	13441.

By repeating the EMFC Program using the results from the Solar Analysis program for the 100, 200, 400 and 800 ft^2 collectors, we obtain economic maximum first costs of $4,105; $7,121; $11,025; $15,005, respectively.

G. Optimal Collector Size

To determine the optimal size from a cost standpoint, the above values are plotted on a curve of economic maximum first cost as a function of collector area as shown in Figure 11-2. The other curve shown on Figure 11-2 is a curve showing the cost of an installed solar system as a function of collector area. In this case, the cost equation has been taken as:

$$\text{Cost} = (\$13.50/ft^2)*\text{Area} + \$3,500$$

This equation gives the storage and collector costs as a function of collector area and includes $3,500 for additional fixed costs. This has been found to be representative of costs for systems installed in the midwest and west. (Note: This figure should be adjusted for local conditions; i.e., in an area of high labor costs, this should be adjusted upward). The intersections of the two curves give breakeven points. That is, a solar system having approximately 110 ft^2 or 840 ft^2 would cost the same over 20 years, as would conventional heating with electricity. The shaded region between the two breakeven points represents a region in which the solar system would have an economic advantage over the conventional system. The point in this domain at which the two curves have maximum separation would represent the optimal design point. This would be at approximately 400 ft^2 of collector area. It is apparent that the separation is fairly uniform over a wide range of collector areas and, therefore, the collector size is not extremely critical in this example. It is possible for the two curves not to intersect anywhere (depending on economic and climatological parameters). In such a case, there would be no economically viable solar design.

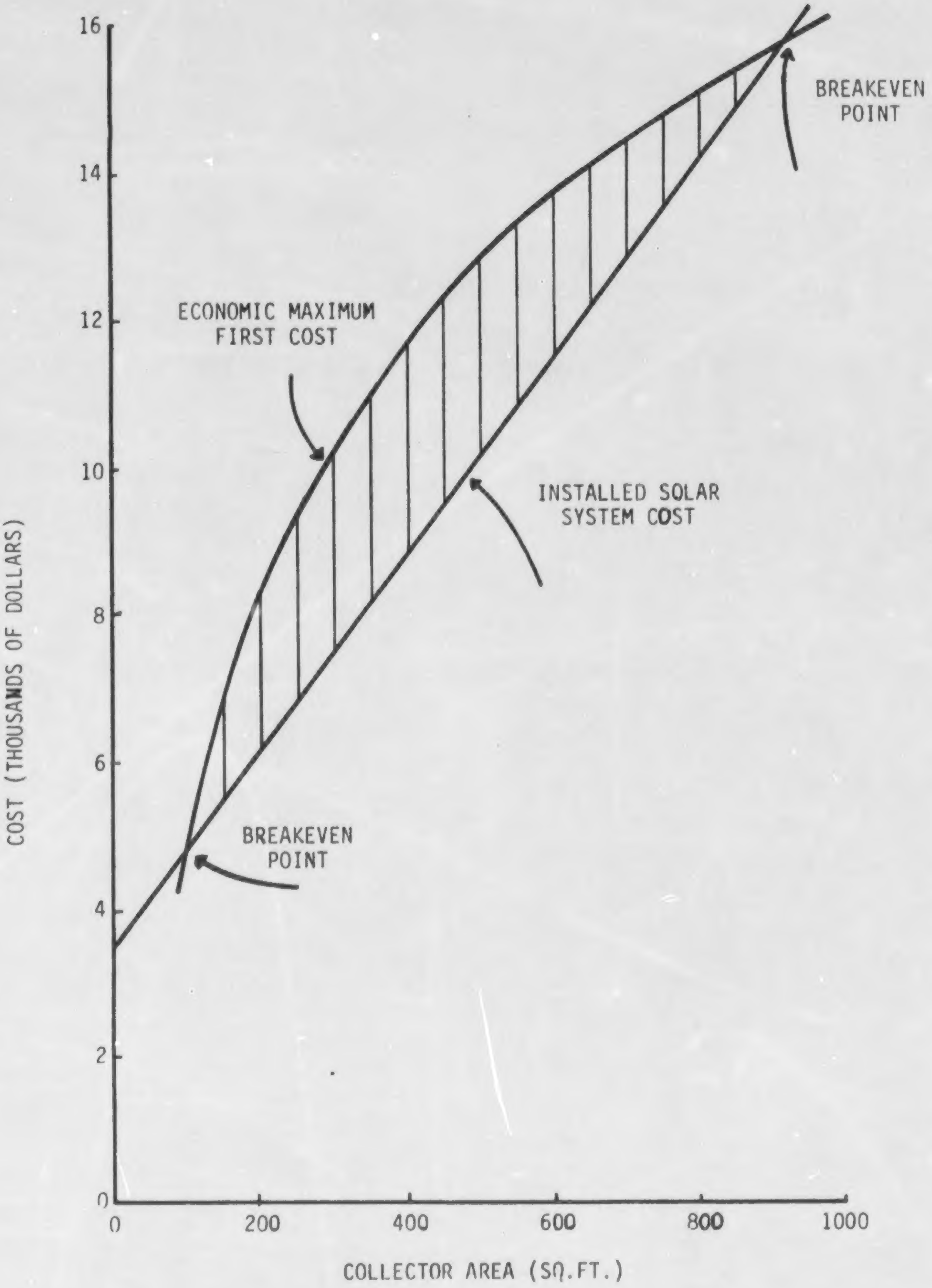


Figure 11-2: Cost Versus Area

There is, of course, more to the design of a solar system than merely sizing the collector array. However, the collector sizing is the first step in the process and is the most significant from the standpoint of cost considerations. The programs that have been presented in this report are very useful in the preliminary design process and have been found to be extremely valuable when discussing the installation of a solar system with a potential client. The fact that answers can be given immediately to questions relating to sizing and economics is very worthwhile.

The remaining system components may be sized by standard design practices in the HVAC industry. For example, for an air system, the blowers should be sized after calculating pressure drops through the storage collectors, and ducting at the flow rates recommended by the collector manufacturer. For air systems, the storage is normally sized at approximately 0.5 ft^3 of rocks per square foot of collector (assuming a pebble bed storage is used). For water systems, a reasonable value is 1.5 gallons of water storage per square foot of collector. Flow rates should be about 2 CFM/ft^2 of collector and 0.03 GPM/ft^2 of collector for air and water systems, respectively.

SIZING, DESIGN AND RETROFIT

SWIMMING POOLS, SPAS AND HOT TUBS

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

SWIMMING POOLS, SPAS AND HOT TUBS

SWIMMING POOL HEATING

Outdoor Pools

Outdoor pools can be heated with unglazed (bare) collectors capable of extending the swimming season by six to eight weeks in both spring and fall. Such a system, while often costing less than one-fourth the cost of the pool itself, in many areas makes the pool useable for about twice the season. Most people consider pool blankets too much of a bother to use for very long, but they will add another month of use to the pool after the solar panels are no longer able to maintain the desired pool temperature.

Some solar swimming pool collectors are made from plastic. While they work well and cost a little less than other collectors, they are less durable. Many of them have become defective in three to five years, thus needing replacement. Others are made from EDPM rubber tubing. These last longer, but because of the poor thermal conductivity of the rubber, their efficiency is low. Other collectors are all metal, with copper water paths, and are glazed. They cost a little more initially, but, with a little care, can be expected to last for the life of the pool.

Unglazed pool collectors are usually mounted directly on the roof, be it sloped or flat. They are relatively inexpensive, and it is often more economical to add an extra panel or two (for optimum heating performance) than to build a frame to support the collectors at a greater tilt angle. Moreover, since the collectors are used primarily in spring and fall, there is no need to optimize winter sun angles.

When mounting the collectors on a flat roof, it is convenient to fasten them with cedar or redwood beams spaced along the top and bottom of the collectors. Use a 4" (wide) x 6" vertical beam at the top and a 4" x 4" beam at the bottom to give adequate drain slope. Box in the sides to prevent wind from entering beneath the collectors.

If the collectors have a metal mounting frame, this frame can be attached directly to a sloping roof with #10 x 2" metal screws. Drill a lead-hole into the roof, fill in and around the hole with silicone sealant, insert the screw and tighten. Apply more sealant over the screw head.

The following chart can be used for sizing collectors (glazed or unglazed) for outdoor pools. There will be some variation, depending on exposure of the pool to wind, night sky radiation, etc., but this simple guide works surprisingly well.

COLLECTOR SIZING GUIDE FOR OUTDOOR POOLS

Collector Slope	Panel Areas as % of Pool Area	
South (slope angle-latitude is best)	50% - 65%	(Use the larger percentage if pool is partly shaded.)
Southeast or southwest	50% - 70%	
Almost flat	60% - 75%	
West	65% - 80%	

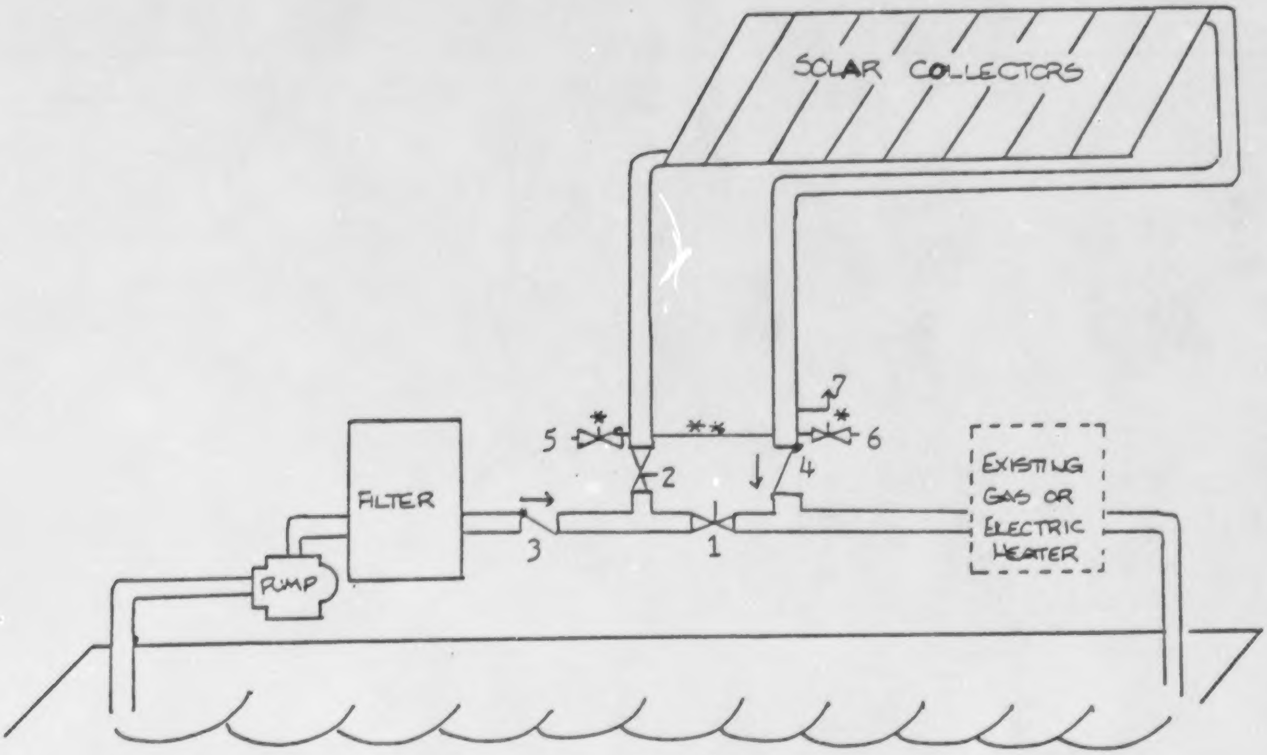
This area of collectors will raise the temperature of the water about 10° - 20° above what it would otherwise be at that time of the year. (Additional collectors can be used, but diminishing returns soon limit their usefulness because of the greatly increased heat losses from the pool as the temperature of the water increases.) The standard pool filter pump is used to pump water through the collector, so no additional pump is required. Instead, the collectors are tied into the return line from the filter to the pool (see Figure 12-1 on the following page).

CONTROL OPTIONS FOR SWIMMING POOL SOLAR SYSTEMS

Method 1: Time switch for the motor. Valves 1 and 2 are manually operated for seasonal change-over (for systems without gas or electric heater).

Method 2: Differential thermostat control for the motor - otherwise, same as Method 1.

Method 3: Differential thermostat control for valves 1 and 2. Motor on time switch if desired. (Recommended for systems with a gas or electric heater.)



Note: 1. Drill a 3/16" hole through valve 1 to permit automatic drain-back at night (not for electric valves).
*2. Seasonal drains for winterizing.
**3. Collector drain tube - used with electric valves only.

Figure 12-1: Solar Pool Heater

Method 1, besides being the simplest and lowest cost, is virtually maintenance-free. It gives good results, but does not optimize each day's collection of solar energy. Method 2 optimizes the collection of energy, but in cloudy weather it may require manual change-over to get adequate filtering. It is medium priced and requires little maintenance. Method 3 optimizes the collection of energy and requires very little attention from the pool attendant, but it is expensive and, worst of all, most of the electric valves need to be cleaned at least once a year for proper operation.

Valve 1 (Figure 12-1) is a by-pass valve, which closes to force water through the collector. Valve 2 is closed when valve 1 is open to prevent circulation through the collectors. Valve 4 is a check valve to prevent water from entering the collectors from the reverse side. Valve 3 is a check valve to prevent back-washing the filter (and losing prime in some systems) when the pump is off. Valves 5 and 6 are small drain valves usually left open in winter when valve 2 is closed. Valve 7 is a vacuum relief valve to let air into the system so the collector will drain. With an electric valve system, it should be as low to the pool as practical, yet high enough to be above the level at which water will stand when valve 2 is closed and valve 1 is open. (Connect the pipes on an electric valve system just above valves 2 and 4 with a 1/4" O.D. tube, thus permitting the collectors to drain.) PVC pipe is normally used for solar pool heating systems, but copper, polyethylene, or polybutylene pipe may be used. (The two-inch size is common for residential size pools.) It is important, of course, that the collectors and pipes be installed so that they will drain back to the pool. If an automatic chlorinator is used, it should be a pressure (rather than suction) type, which puts the chlorine into the water after it has left the collector. If copper piping is used, be sure to monitor the pH level of the water or the chlorine may attack the copper, resulting in severe corrosion.

Indoor Pools and Hot Tubs

Indoor pools can also be heated effectively with unglazed panels, except during the winter (when the average daytime temperature is more than 20° below the desired pool temperature).

The sizing guide for outdoor pools given earlier (figures for partly shaded pools) may also be used as a rough sizing guide for indoor pools. For more accurate sizing, compute the average heat loss from the pool in conjunction with the gain from the collector(s). (Temperature losses from swimming pools can be calculated by the formula in the following pages).

When this formula is applied to indoor pools, it will be apparent that heat losses in mid-winter are quite substantial. Thus to heat an indoor pool through the winter will not only require covered collectors (single-glazed) but plenty of them. Mid-winter heat losses, however, can be considerably reduced by covering the pool except when it is being used. With private pools, the surface can be covered except for a "swimming lane" five feet wide down the length of the pool. If the pool temperature is to be maintained through the winter, it will often be more economical to use an auxiliary heater to help with the winter peak load.

Hot tubs or spas can also be heated with solar collectors, with the tub being covered except when in use. A single-glazed collector of 32 to 64 sq ft is usually adequate. An above-ground spa will probably require insulation on the outside of the tub to keep heat losses to reasonable levels. For this system, it is often more desirable to use a separate pump, controlled by a differential thermostat, than to use the filter pump.

Pool Temperature and Heating Requirements

A pool temperature of approximately 85° F (about 15° below body temperature) is considered ideal, though generally acceptable pool temperatures may range from 70° (minimum) to 93° (maximum).

Heat losses from swimming pools result primarily from two major factors: evaporation and convection at the surface. Radiation is a contributing but not a major factor, and conduction losses into the surrounding earth are significant only during the initial heating of the pool (usually in the spring). Both convection and evaporation losses are directly related to the difference between pool temperature and ambient temperature, as well as prevailing wind velocity, with evaporation losses tending to increase particularly as the ambient wet bulb temperature decreases. Figure 12-2 indicates generally how swimming pool losses and collectable solar energy may vary with the season. Note the potential extension of the swimming season with solar collectors. It should further be noted that heat losses from indoor pools will be less severe than those suggested in Figure 12-2, on the following page, depending on the particular enclosure for the pool. In fact, the swimming season may be extended year-round by solar-heating an indoor pool, though a considerable collector area may still be required.

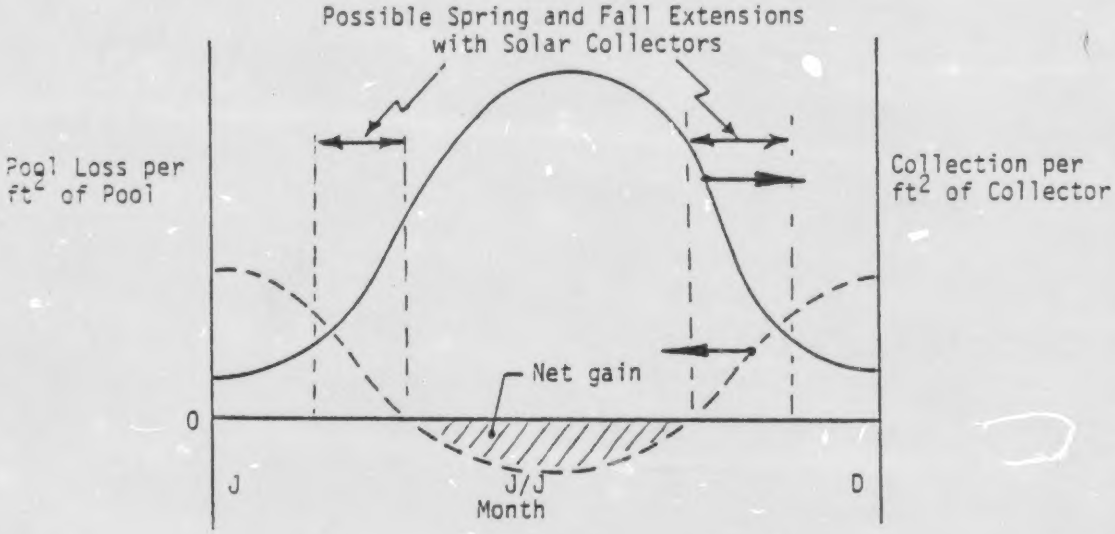


Figure 12-2: Pool Losses and Potential Solar Collection

Auxiliary/Back-up/Passive Systems

Auxiliary pool heaters are usually gas-fired, as opposed to electric, because electricity for pool heating is not only expensive but generally wasteful of high-grade energy. As stated above, the gas-fired auxiliary heater is ordinarily installed in the return between collector and pool.

Swimming pool solar heating systems are occasionally integrated with residential heat pump systems. In the winter months, heat is pumped from the pool to the house, while in summer, during the air-conditioning season, unwanted heat energy is rejected from the house to the pool. Although such a system may well facilitate winter heat-pump operation, it may also reduce the swimming season, depending on the size of the solar collector array. Furthermore, during the air-conditioning season, the energy rejection to the pool from the cooling condenser coil may result in excessively high pool temperatures, which may require nocturnal circulation, thus negating the benefits.

Finally, the swimming season for a pool may be significantly extended by "passive" solar methods. One very effective method is to cover the pool with sheet plastic or commercially available plastic "pillow" material. Sheet plastic covering the pool will effectively eliminate the evaporation loss, which is particularly critical in the cold-dry season. While these materials must be manually removed and periodically replaced, they are a relatively inexpensive and effective means of extending the swimming season.

Pool System Sizing

The following several pages will present data unique to pool systems. The sizing process for components is the same as other heating applications, except in the areas outlined hereafter.

The following table (Figure 12-3) outlines various swimming pool temperature requirements.

short swim periods	almost any temp.
after a sauna bath	almost any temp.
competitive swim pool	72° F
recreational swim pool	80° F
early morning swim	85° F
evening swim	85° F
small children	85° F

Figure 12-3: Pool Temperature Requirements

Heating a swimming pool is an ideal use of solar energy. The heat is needed at low temperatures, so that simple collector designs can be used. A pool is equipped with a filter and circulating pump, so the solar collector is generally supplied with a flow of pre-filtered, chemically treated water. Temporarily bad weather is not too bothersome; the effectiveness of solar heating is reduced, but swimming desires normally fluctuate in phase with the weather.

In a well-filtered pool with an average depth of about 5 feet, at least 75% of the incident solar energy will be absorbed by the water. Approximately 5% of the solar energy is reflected from the surface water, while the remaining 20% is able to penetrate the bottom or sides of the pool, reflecting back through the water and away from the pool.

The heating requirements for a swimming pool can be obtained by simply calculating the heat loss of the pool at the desired operating temperature. Part of this heating requirement, as suggested above, is received free of charge from direct sunlight, and the rest must be supplied externally.

There are a number of factors involved in the loss of heat from a pool. Normally the losses into the ground are both negligible and readily returned whenever the pool temperature drops. In addition to convective heat transfer losses to the air, the most significant losses from swimming pools are evaporative losses.

The equation below gives the total heat loss per hour due to all of these factors. The evaporative heat loss has been included, as suggested earlier, by using the convection heat transfer coefficient, water vapor pressures, and a proportionality factor.

The equation is:

$$Q_L = A_p \, [ER + 1.0\,(t_w - t_a) + h_{ca}(t_w - t_a) + 200h_{ca}\,(P_w - P_a)]$$

where

Q_L is the heat loss from the pool in BTU/hr

A_p is the pool area, ft^2

E is the effective emmissivity of water, 0.9

R is the radiation heat loss to the sky of a black body ($E = 1.0$) at t_a (about 25 $BTU/ft^2/hr$)

t_w is the pool temperature, ^oF

t_a is the ambient temperature, ^oF

h_{ca} is the convection heat transfer coefficient between the pool water and the air, equal to 1.5 $BTU/ft^2/hr/^oF$ at 6 mph wind speed (see Figure 12-4 for other wind speeds)

P_w is the water vapor pressure in equilibrium with the swimming pool water as found in Figure 12-5, lb/in^2

P_a is the water vapor pressure in the air, as found from Figure 12-4, lb/in^2

The water vapor pressure P_w in equilibrium with the pool water at temperature t_w can be found from Figure 12-5 by using the 100% relative humidity line, and using t_w for the temperature. P_a can be determined from Figure 12-5 with input values from daily weather reports.

The equation can be modified to determine the change in heat loss, Q_L, as a function of pool temperature change. It can be found from Figure 12-5 (the 100% relative humidity curve) that for every degree (°F) change in the pool temperature, the water vapor pressure P_a goes up roughly 0.016 lb/in^2. We may then develop from the previous equation

$$\Delta Q_L = A_p\,(1 + h_{ca} + 3.2h_{ca})\,(\Delta t_w).$$

$$\Delta Q_L = A_p\,(1 + 4.2h_{ca})\,(\Delta t_w).$$

Only the terms involving t_w in the first equation are used in the second equation. The product of the factor 200 in the first equation and the factor of 0.016 leads to the factor of 3.2 in the last (evaporative) heat loss term in the second equation. It is clear that as the pool temperature is increased, 3.2 times more heat is lost by evaporative cooling than by convective cooling. This is why a pool cover is so effective, although its sole function is to prevent evaporation.

If a pool temperature analysis for a particular time of year is deemed necessary, the system designer need only equate the solar radiation collected in a given period to the energy required to raise the pool temperature.

POOL HEATING COLLECTORS

Many types of collectors are suitable for pool heating. The temperature difference between the water to be heated and the surrounding air is small, so expensive design features to reduce collector heat loss are not required.

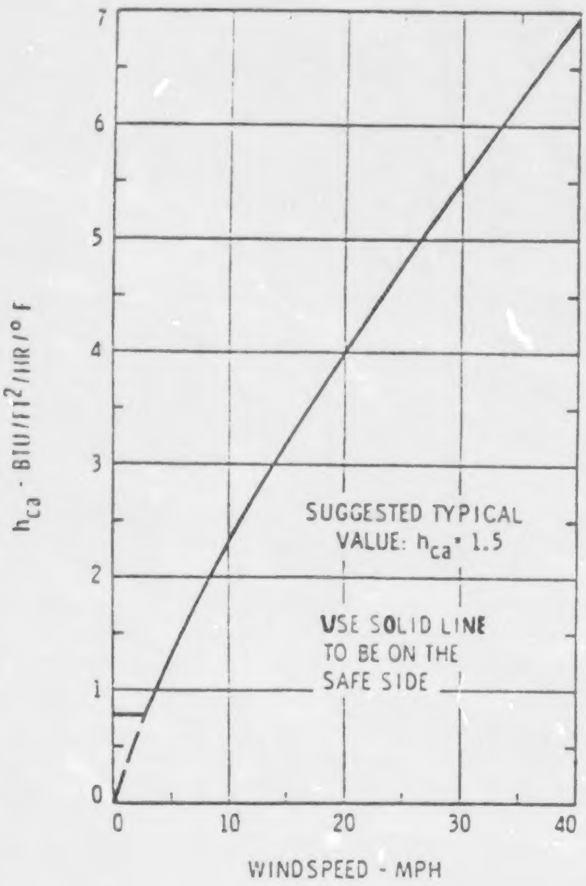


Fig. 12-4: How Convection Heat Transfer Coefficient h_{ca} varies as a function of Windspeed.

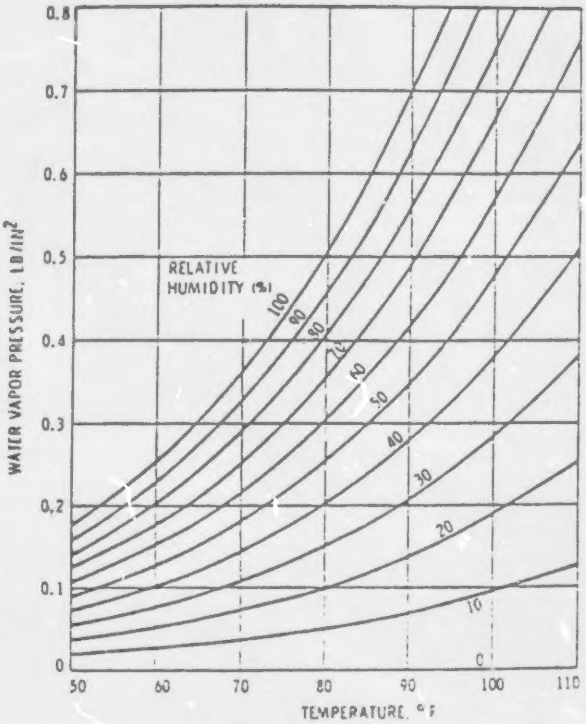


Fig. 12-5: Relationship Between Water Vapor Pressure, Air Temperature and Relative Humidity

Performance of typical pool heating collectors is shown in Figure 12-6, which offers a wealth of information. Efficiency, defined as the amount of energy collected divided by the total amount of solar energy striking the collector surface, is displayed on the vertical axis. (One of the goals in selecting solar collectors is to obtain the highest efficiency at the least cost.) The horizontal axis shows the fluid parameter, which involves three variables: collector temperature (T_c), air temperature (T_a) and amount of solar radiation striking each square foot of area (I). By conveniently defining the fluid parameter to be $\frac{T_c - T_a}{I}$, the collector efficiency graphs are smooth curves which are nearly straight lines.

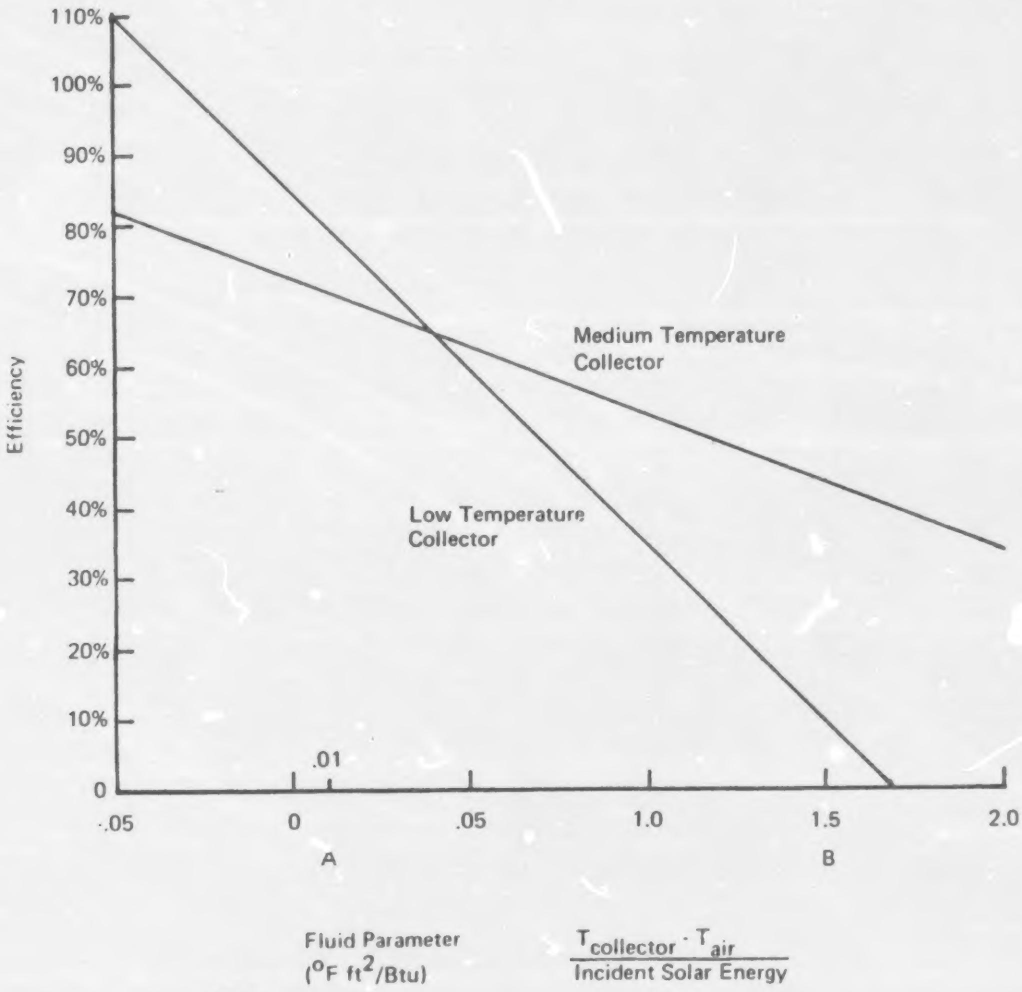


Figure 12-6: Collector Efficiency

Note that for both types of collectors efficiency decreases as the fluid parameter increases. Factors which reduce collector efficiency are: decreases in air temperature, decreases in solar radiation and increases in pool temperature.

The following two examples are marked in Figure 12-6.

	A	B
Pool Temperature	78° F	80° F
Air Temperature	75° F	65° F
Solar Radiation	$300\ \text{Btu/ft}^2\text{-hr}$ (sunny)	$100\ \text{Btu/ft}^2\text{-hr}$ (cloudy)
Fluid Parameter	$\frac{78\text{-}75^\circ\ \text{F}}{300\ \text{Btu/ft}^2\text{-hr}} = 0.01$	$\frac{80\text{-}64^\circ\ \text{F}}{100\ \text{Btu/ft}^2\text{-hr}} = 0.15$

Case A represents conditions on a relatively warm, sunny day. In this situation, a typical low-temperature collector is more efficient than a typical medium-temperature collector and would capture more energy per square foot. During a cool, cloudy period, such as that represented by case B, the situation is reversed and the medium-temperature collector is the more efficient.

In pool heating applications, air temperature frequently exceeds pool water temperature and energy can be captured directly from the air. For these situations, the fluid parameter is negative, and efficiencies greater than 100% may result. Low-temperature collectors are ideally suited to take advantage of this effect.

Low-Temperature Collectors

The most common types of low-temperature collectors are flat-plates and pipes.

Flat-Plate Collectors

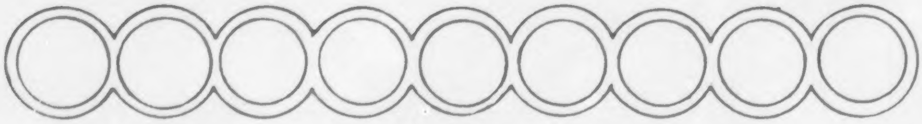
Several types of flat-plate collectors specifically designed for pool heating are available in both plastic and metal, plastic being the least expensive and most popular.

Flat-plate collectors are characterized by large-diameter headers at each end and numerous small fluid passageways along the plate portion. The header's primary function is to distribute the flow evenly to the small passageways connected to it. If the header is large enough, it can also serve as distribution piping, which reduces material and installation labor costs. The fluid passageways, which collect energy from the entire surface, are small and distributed across the plate so that a maximum amount of the collector area is wetted.

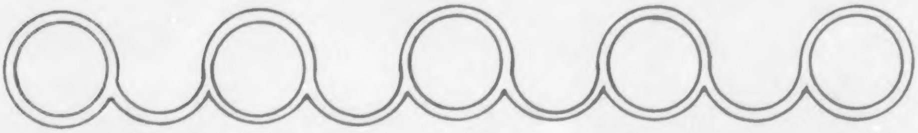
Plastic collectors must be well designed to capitalize on the unique features of plastic and to avoid some of the material's shortcomings. Plastic is a poor conductor; consequently, flow channels must be closely packed, and material thickness must be kept to a minimum to reduce the temperature difference between collector surface and circulating water. If the collector surface becomes excessively warm, a large amount of heat energy can be lost to the atmosphere. Designs by two large plastic collector manufacturers are shown in Figure 12-7.

High water flow rates also keep collector temperatures low. The total amount of energy (the most important variable) delivered to the pool is the product of the amount of water flowing multiplied by its temperature rise. (Five hundred gallons of water raised 1° F contains as much energy as 10 gallons of water raised 50° F, but the collector operating at 1° F above pool temperature will operate more efficiently.) Increasing collector efficiency by increasing the flow rate through it is desirable. One gallon per minute for each 10 square feet of collector area is frequently recommended by many manufacturers.

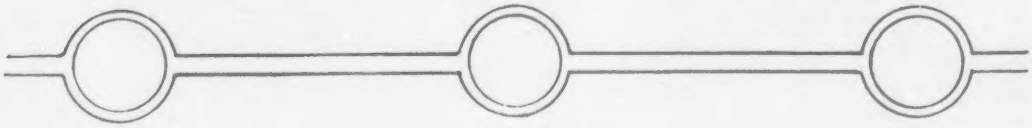
Plastics are available in numerous formulations and types, many of which are immune to attack from common chemicals, but most plastics degrade in sunlight. Of the myriad possibilities, polypropylene, acrylonitrile-butadiene-styrene (ABS), polyethylene, polybutylene and polyvinylchloride (PVC) seem to be the most frequently used collector materials. They have been used in pool equipment for several years and have demonstrated their ability to withstand swimming pool chemicals.



PLASTIC



PLASTIC



METAL

Figure 12-7: Typical Collector Designs

Plastic used in a solar collector must withstand years of exposure to sunlight. The untraviolet portion of sunlight can break chemical bonds in most plastics and eventually destroy the material if the process is not prevented. Carbon black, added in the amount of three to five percent, absorbs most of this damaging radiation and retards degradation in addition to improving the collector's absorptivity. Other additives are available, and collector manufacturers use several proprietary combinations. The complexity of the degradation process makes estimation of material lifetimes difficult; therefore, explicit, extended warranties are desirable, and most manufacturers currently offer either a five-year or ten-year warranty.

Flat-plate collector designs utilizing metals are slightly different from plastic configurations. Metal being a superior conductor, relatively long fins can separate the tubes without causing excessive operating temperatures on the collector (Figure 12-7). In fact, metal absorber plates such as those used in glazed water heating collectors frequently serve for pool heating. Metal collectors that are wetted continuously on the backside of the collection surface are slightly more efficient than fin-type collectors, but are also more expensive.

Pipe Collectors

A large array of plastic pipe can be used as a swimming pool heater. Lengths of pipe may be placed adjacent to one another, with a small gap between each, until enough collector surface has been laid down to capture the necessary solar energy - the pipes themselves are the solar absorbers.

Careful attention must be paid to materials. Many types of plastic pipe are available, and ABS (acrylonitrile-butadiene-styrene) with proper ultraviolet inhibitors seems to have performed well in Florida to date. Compatible fittings, solvent and cement should be used to solvent-weld all joints to ensure durable, watertight connections.

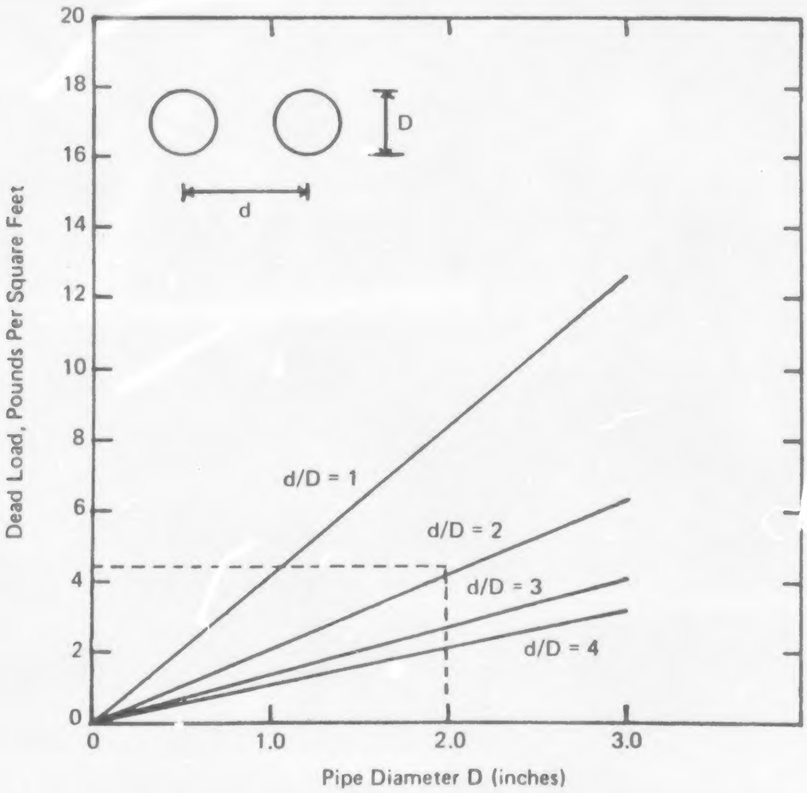


Figure 12-8: Dead Loads of Pipe Collectors

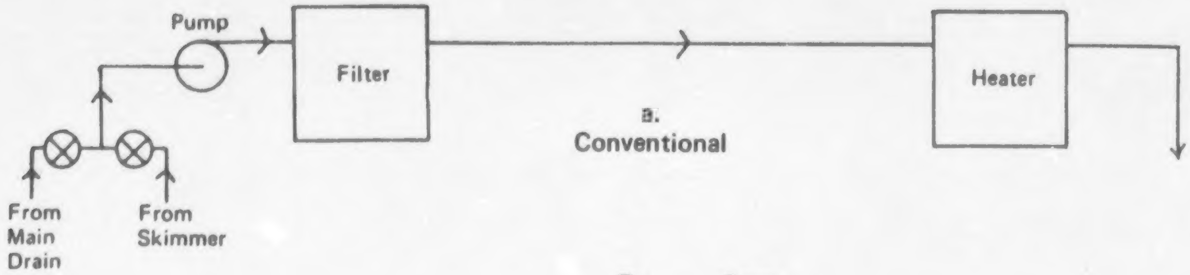
Pipe collectors are convenient because sizing is flexible and the pipe is easy to install, but excessive roof leads must be avoided. The total dead load of the pipe collector when it is filled with water depends upon the diameter of the pipes and the spacing between adjacent pipes, as shown in Figure 12-8, on the preceding page. For example, 2-inch diameter pipe, placed two diameters apart imposes a dead load of 4.2 pounds per square foot. Most roofs should support this additional load, but every case should be examined to ensure conformance with local building codes.

Flow Control Devices

Solar pool heaters are generally connected to existing pool plumbing systems. This section explains how to make the connections. A schematic of the most common pool plumbing is shown in Figure 12-9a. The pump draws the water from the skimmer and main drain, forces it through the filter and returns it back to the pool through the conventional heater. Occasionally, skim filters will be encountered where the filter **is** ahead of the pump. Connections for the solar system are identical for both plumbing arrangements so they will be shown on the most frequently encountered configurations.

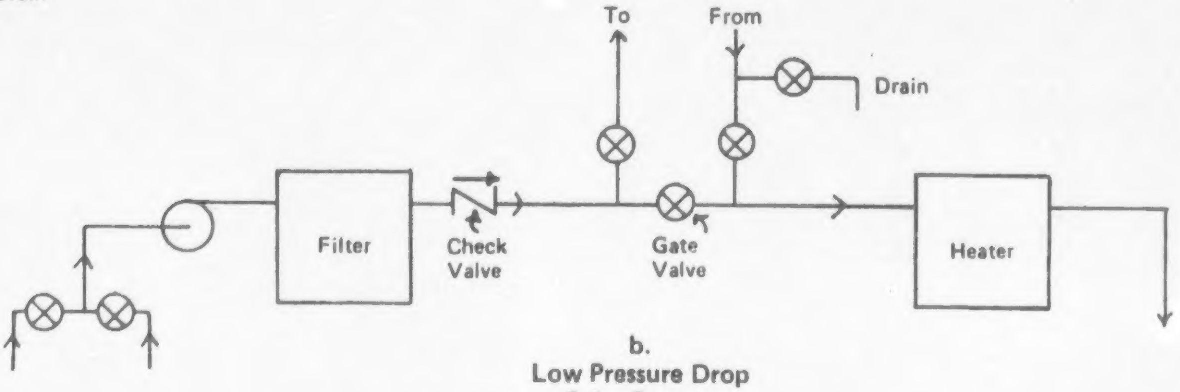
Solar systems designed to operate with small-pressure losses can be plumbed as shown in Figure 12-9b. A spring loaded check valve is installed downstream from the filter to prevent water from the collector array from backwashing through the filter when the pump is shut down. A manually-operated gate valve is placed in the main line between T's that feed the collector bank and return the solar heated water. Gate valves also are placed in the feed and return lines for insolating the solar system from the pool filtration system when the filter is being backwashed or if adjustments must be made to the solar system. When solar heating is desired, the pump timer is adjusted to operate during daylight hours, and the gate valve in the mainline is manually closed slightly to throttle the flow and force water up through the collectors. Gate valves on the lines to and from the solar system should be fully open.

Flow through the collectors should be increased by closing the throttling valve until return water from the collectors feels approximately the same



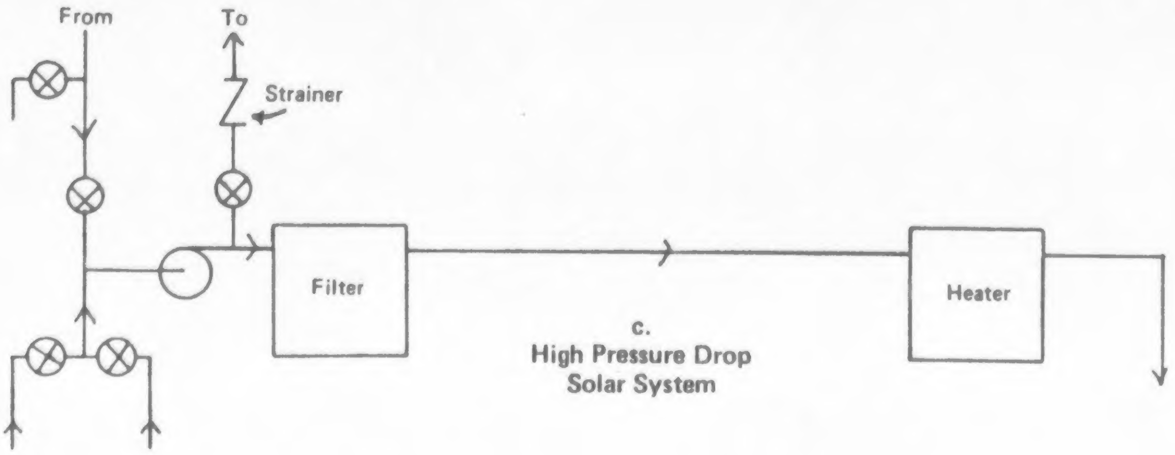
a.
Conventional

Pump
Filter
Heater
From
Main
Drain
From
Skimmer



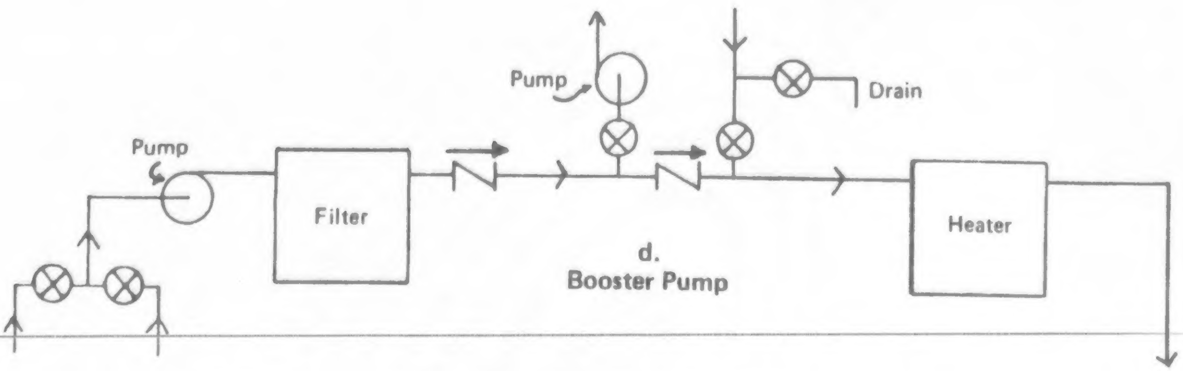
b.
Low Pressure Drop
Solar System

To
From
Drain
Filter
Check
Valve
Gate
Valve
Heater



c.
High Pressure Drop
Solar System

From
To
Strainer
Filter
Heater



d.
Booster Pump

Pump
Drain
Pump
Filter
Heater

Figure 12-9: Plumbing Schematics

temperature as the feed water. It may seem logical to reduce the flow rate through the solar array to make the return water warmer, and this can be done; however, it is not logical - the collectors will be forced to operate at higher temperatures, their efficiencies will drop, and less solar energy will be delivered to the pool. The temperature rise through the collectors should be kept low - approximately 6° F or less on warm, sunny days.

Forcing water through the solar system uses some of the pump's power, thus reducing flow through the pool filtration system. As the throttling valve is closed, pressure on a gauge mounted on the filter or discharge side of the pump will rise slightly, approximately 3 psi or less for a well-designed residential system. If the throttling valve were closed entirely, all of the flow would be diverted to the solar array, and its efficiency would increase. If the pressure at the filter does not rise unduly, the solar system can accommodate the total flow. If the pressure rises over roughly 5 psi, flow through the filtration system may be restricted too much, and the throttling valve should be opened slightly to allow some of the flow to bypass the collectors. An inexpensive plastic flow meter can be used on the main line after the collector return connection to monitor flow rates through the filtration system. Check with local building officials to determing minimum filtration flow rates or pool turnover times required in your area.

Solar systems that require high-pressure drops to obtain sufficient flow can be installed with extra pumps or plumbed as shown in Figure 12-9c. The feed line to the collectors is connected to the discharge side of the pump, ahead of the filter, and the return line is connected to the suction side of the pump. The full pressure drop across the pump is available to force circulation through the collectors.

When solar heating is desired, gate valves to the solar system are manually opened. If the pressure at the filter gauge drops substantially, or a flow meter on the main filtration line indicates that the pump is being short-circuited by excessive flow through the collector loop, one of the valves can be closed slightly to divert more water through the filtration system.

Since water will pass through the collectors prior to the filter system, a strainer is recommended on the feed line to capture medium-size particles that may clog small fluid passageways in the collectors.

When the existing pool pump lacks enough power to circulate sufficient flow through the solar system and the filtration system in either of the configurations shown, a second pump may be required - it should be installed as shown in Figure 12-9d. Common pool-circulating pumps without the filter basket are suitable for this application.

The solar pump should be placed in the line feeding the solar collectors, not in series with the filtration pump. An additional check valve may be required in some cases to prevent undue recirculation if the filter pump is especially weak.

The solar pump may be operated by the same time clock as that for the filter pump, or it may have a separate control. If both pumps operate from the same timer, it should be set so that the pumps come on during daylight hours. If the solar pump is separately controlled, the filter pump may run for a longer portion of the day, and the solar pump should turn on during daylight hours only when the filter pump is operating.

Manual flow control or control with time clocks is simple and inexpensive but has drawbacks. Since clocks do not respond to weather conditions, only to time, the circulating pump may be operating when there is insufficient solar energy available. Collectors occasionally will lose energy rather than gain it if weather conditions are unfavorable, especially if they are low-temperature collectors. Manual control overcomes this limitation but requires more personal attention than many pool owners wish to give.

Automatic flow controls are available from several manufacturers. The most common plumbing schematic for systems using these devices is shown in Figure 12-10, on the following page.

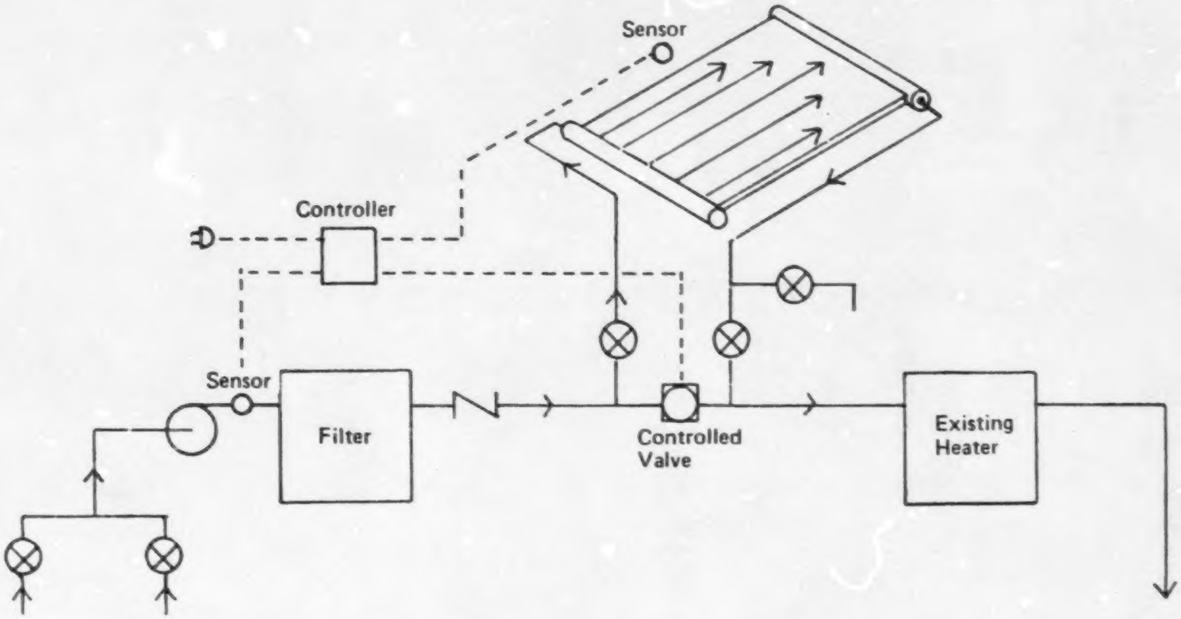


Figure 12-10: Automatic Plumbing Control Schematic

Accurate differential temperature controls yield good overall solar system performance. A sensor, tapped into the piping at a convenient place ahead of the collector return line, measures the pool water temperature. Another sensor is housed in a plastic block and placed near the solar collectors so that its temperature simulates heat of the collector. When the water temperature exceeds the simulated collector temperature, the control valve remains in the open position, and the flow bypasses the collector loop. When the water temperature is below the simulated collector temperature, useful energy can be extracted from the solar system - the valve is closed, forcing flow through the collectors and warming the pool.

The electronic control automatically adjusts to changing conditions, monitoring variations in collector temperature caused by clouds, other weather factors, or the approach of evening.

When collector temperature drops, the control de-energizes the valve and flow again bypasses the collector. Maximum pool temperature limits can be programmed into some controls.

Automatic controls increase efficiency of solar heating systems and make them more convenient to operate. Low-temperature collectors, when operated on a cool, rainy day, can lose energy. By preventing this occurrence, controls assist in the maintenance of warmer pool temperatures. Properly-operating systems require only seasonal adjustment and little or no attention otherwise.

Control valves may be operated by hydraulic or electrical actuators. One of the earliest valves used, and one that is still popular today, is a hydraulically-operated pinch valve consisting of a sturdy rubber cylinder with expandable bladder inside. A high-pressure line connected to the discharge side of the pump is used to expand the bladder, pinching off the flow and diverting it through the solar system. A low pressure line connected to the suction side of the pump deflates the bladder and allows the flow to pass unimpeded. Switching between the high and low pressure lines is accomplished by an automatic control.

Electrically-operated valves also are common. The control signal may be used to operate a small solenoid that in turn activates the main valve. Operation may be assisted by a pressure line from the pump. Originally, irrigation valves were used for this purpose, but pressure drops across the valves were excessive. Now, specifically designed and constructed valves are available.

Corrosion Problems

From experience, metals can be listed in a galvanic series, which can be useful in predicting which metals are acceptable for use in contact with one another and which materials are likely to be corroded. Following is a simplified galvanic table.

CORRODED END - ANODIC - LEAST NOBLE

- Magnesium
  - Zinc, Galvanized Steel
    - Aluminum
      - Mild Steel, Cast Iron
        - Lead, Tin
          - *Brass, Copper, Bronze
            - *Nickel-Silver, Copper-Nickel Alloys
              - *Monel
                - Stainless Steels

PROTECTED END - CATHODIC - MOST NOBLE

Material groups marked with an asterisk have no strong tendency to produce galvanic action and from a practical standpoint are safe to use in contact with one another. However, the coupling of two metals from different groups will result in accelerated corrosion of the metal higher in the series. The farther the metals are apart in the series, the greater the galvanic corrosion tendency.

The three dominant factors affecting galvanic corrosion are: (a) potential difference between the dissimilar metals, which is function of the distance apart of the metals in the galvanic series; (b) conductivity of the water, which increases as the ionic concentration in the pool water increases; and (c) relative area of the two metals forming the couple. Selecting compatible materials and maintaining acceptable pool chemistry affect the first two factors. Corrosion is directly proportional to the ratio of the area of the protected or cathodic material and that of the corroded or anodic material. Avoid using combinations where the area of the anodic material is relatively small. In contrast, it is a good practice to use the more noble (cathodic) materials for fastenings and other small critical parts.

A second common form of corrosion, due to concentration cells, is associated with crevices, scales, other deposits, or any other means by which differences in solution concentration or composition occur over a metal surface.

Corrosion will attack areas of weaker concentration more intensely and rapidly than it will uniformly exposed areas. Such cells can be set up by local difference in oxygen, chlorine, temperature, agitation and, in fact, by almost any dissimilarity in exposure conditions.

Erosion-type corrosion may be caused by impingement of high velocity water streams that break through protective corrosion scales or films, thereby exposing new materials to attack. This velocity effect can be combated in pool equipment by avoiding unnecessarily high velocities in piping and by designing to avoid highly turbulent areas. For this reason, it is considered good practice to limit actual flow velocities in copper piping to less than about 7 feet per second and in cupronickel piping to about 12 feet per second.

Improper pool chemistry also can cause accelerated corrosion. The pH of pool water should normally be maintained between 7.4 and 7.8, and no corrosion problems with copper or copper alloys should be expected from pH levels above 7.2. Low pH (less than 7.2) is generally caused by improper addition of pool acid.

COLLECTOR INSTALLATION

Acceptable solar collector mounting practices are discussed in this section. Since low-temperature collectors are the most popular for swimming pool heating, procedures for mounting plastic, flat-plate collectors and pipe arrays will be discussed first.

Please remember - in the construction industry, safety is everyone's responsibility. Installing a solar system is a relatively straightforward job if safe work habits are practiced.

Flat-Plate Collectors

The optimum installation angle is latitude plus 10 degrees, and the collectors should face south if possible. Plastic flat-plate collectors are rather limber and are not self supporting. If suitable roof space is available, the collectors can be mounted directly on the roof - this is the most common configuration. Supports can be constructed to mount collectors at the proper orientation, but, except in new construction, the additional cost is generally prohibitive. The number of collectors should be increased for less than optimum orientations.

Collectors should be securely fastened down to withstand maximum expected windloads. Building code requirements for maximum wind velocities vary from 130 to 80 miles per hour. Loads at heights below 50 feet may exceed 60 pounds per square foot. Check with the local building inspectors for windload provisions in your area.

Numerous mounting techniques are available to meet the requirements of the various collector designs. Test data is generally not available for most configurations, but one of the largest collector manufacturers did run wind tunnel tests on its installation technique, and the collectors withstood winds in excess of 100 mph. That company's recommended installation procedures form an important part of the practices described in this section.

Installation begins with laying out the collectors on the available roof area in such a way that shading by trees, parts of the building, and other obstructions is minimized. If large numbers of collectors are involved, they may have to be divided into several banks, with collectors in each group plumbed in parallel. Plumbing arrangements between banks are discussed in the next section.

Once the panels are placed, they should be fastened together. Short, flexible couplings made of EPDM or butyl rubber are generally used. They slide over the ends of the header and are clamped down firmly with stainless steel, screw-tightened clamps similar to those used on automotive radiator hoses. Once fastened together, the collectors are cumbersome to move about, but it can be done.

Collectors can be readily mounted on asphalt shingle and tar and gravel roofs. Insulating underlayment will protect the panels from abrasive roofing materials and prolong their lifetime. It will also protect the roof from overheating.

The substrate can also provide ventilation for the roofing membrane beneath, which will retard the development of fungus or mildew. Common 2-1/2-inch, corrugated, 4-ounce fiberglass panels are among the least expensive materials available for supporting plastic solar panels and will last indefinitely.

Opinions of experienced roofers vary as to the desirability of the underlayment. Some recommend at least a 1-1/2-inch air gap to provide circulation; others feel that a substrate may not be required at all. Numerous systems have been installed in the West without substrates. Conservative practice favors the use of an underlayment, but the subject has not been resolved.

Collectors should be laid on the roof and fastened properly at the header on either end. At least two, and preferably three, straps should span the panel to prevent it from windlift. Figure 12-11 shows one possible arrangement.

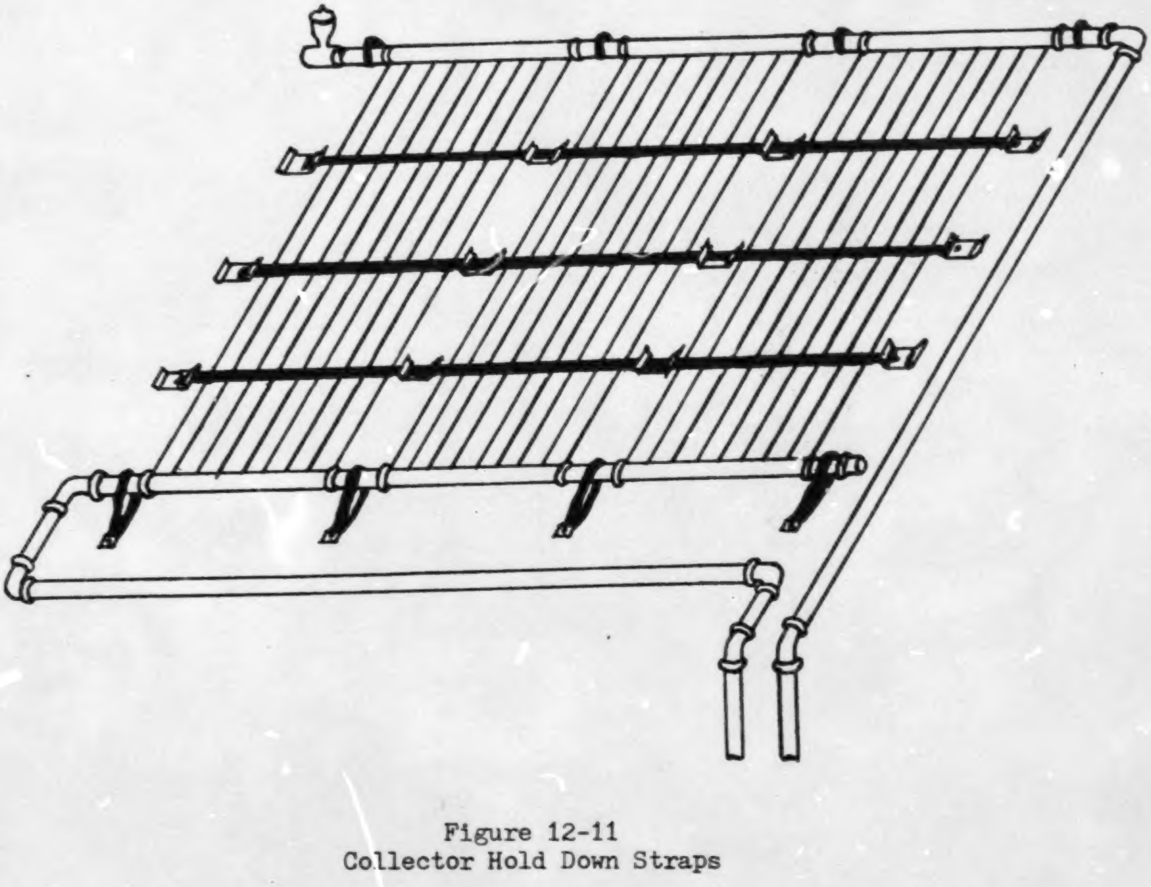


Figure 12-11
Collector Hold Down Straps

One end of the panel can be fastened to the roof with a short strap or clamp around the header. The other end should be fastened with an elastic material or spring to allow for expansion as the collector temperatures change - 10 foot plastic collectors may vary up to an inch in length. Generally the bottom header is pulled down until the upper straps are taut, then the lower straps are stretched out and fastened to clips mounted on the roof. Lower straps usually are four to seven inches long.

Straps should be installed across the panel body; one at either end, within a foot of the headers, and one across the middle are recommended. They should be made of material such as nylon or plastic-coated metal that will not scratch or abrade the collector, since they will lie directly in its surface. The bands should be secured to clips fastened approximately an inch from the collector edge and pulled snug to hold the collector down.

Figure 12-12, on the following page, shows a typical mounting clip, which may be made of rigid plastic or metal. On asphalt shingle roofs, the clips may be fastened directly on top of the shingles - 1/4-inch lag screws long enough to penetrate the roof sheathing are generally recommended. The lag screws should be screwed into roof rafters (rather than just roof sheathing) to protect the collector against hurricane-force winds. Because this mounting procedure is so tedious, many installers follow the simpler practice of driving directly into the sheathing - only time will tell which procedure will become accepted practice. A pilot hole should be drilled for the lag screw, and after the drill chips are brushed away, a liberal amount of sealant should be injected into the hole with a cartridge gun or similar device. Excess sealant should be placed on the surface where it can seal between the clip base and the roofing material when the lag is screwed down tight. Polysulfide or silicone sealants adhere well to common building materials and seem to give good results.

Special care should be exercised in sealing every lag screw where it penetrates the roof membrane. Continuous attention is required to make each fastening strong and watertight.

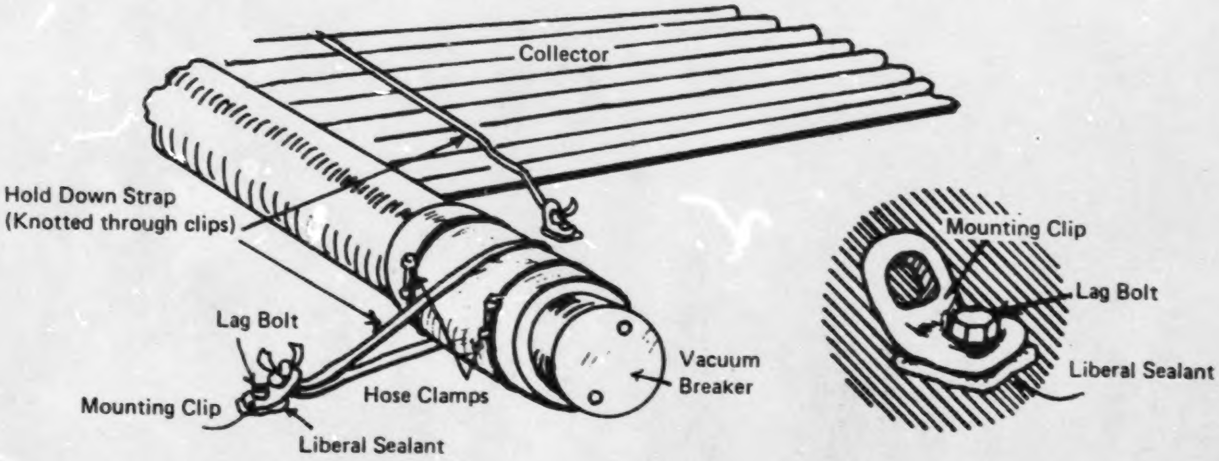

Collector
Hold Down Strap
(Knotted through clips)
Lag Bolt
Mounting Clip
Liberal Sealant
Hose Clamps
Vacuum
Breaker
Mounting Clip
Lag Bolt
Liberal Sealant

Figure 12-12
Typical Collector Mounting Clip

Sealing of mounting brackets on tar and gravel (builtup roofing) required careful cleaning around the bracket. Scrape off the old gravel down to the tar and clear off all dirt and residue. If the tar surface is very dirty or irregular, soften it with a solvent such as mineral spirits. After sealing the clip with polysulfide, pour roofing tar over the bracket base and cover with gravel - this last step is mandatory for installations on flat roofs.

Mounting collectors on other roof types is more difficult. On cedar shake roofs, mounting screws should pass through the shakes and fasten securely to the plywood or stringers beneath. Liberal amounts of sealant are recommended. When drawing the fastener down tight, be careful not to exert excessive pressure on the shake that may cause it to bow or split.

Concrete tile roofs present special mounting difficulties. The best solution is to construct a rack to support the collectors above the tile surface. The rack should be constructed of durable materials such as aluminum, cedar or pressure-treated wood, and able to withstand maximum anticipated windloads. Substrate and collectors are fastened to it.

The rack must be securely fastened to the roof trusses - not to the sheathing. Figure 12-13, on the following page, illustrates a typical mounting bracket arrangement. To install the bracket, a tile must be removed or broken out to reveal the waterproof menbrane on the sheathing below. Note that this heavy tarpaper (commonly called slate), not the tiles, forms the water barrier.

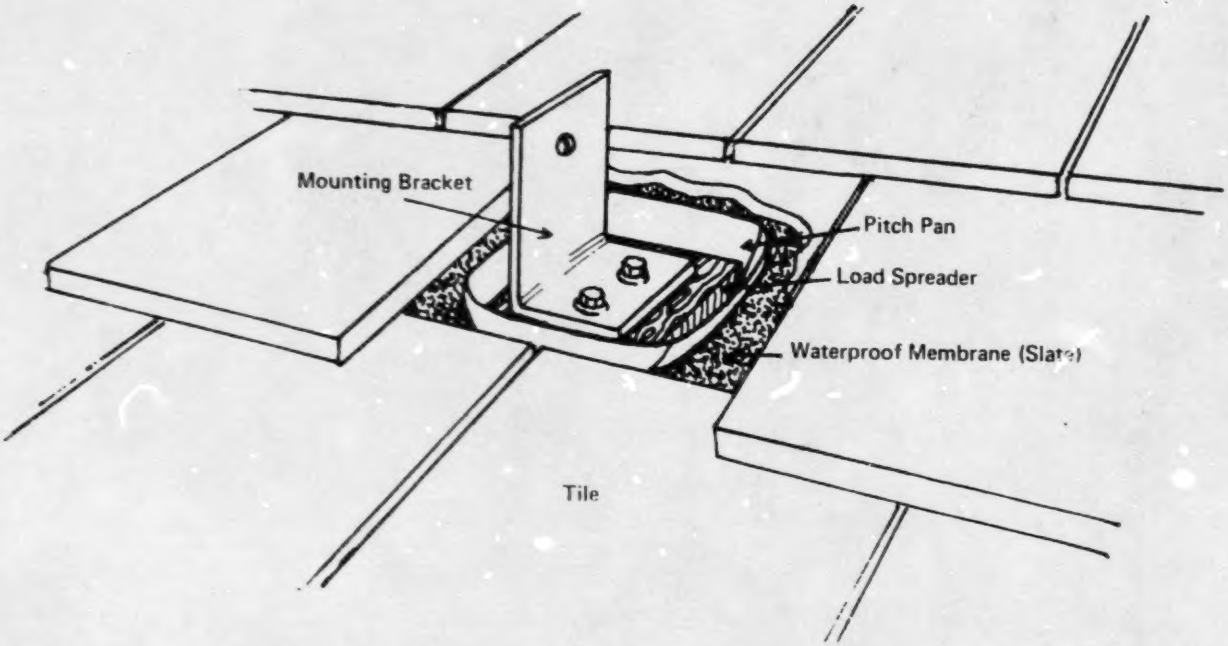


Figure 12-13
Mounting Bracket for Tile Roofs

The mounting location should be swept clean of dust and debris. Roofing mastic should be applied to the bracket base and slate to form a good seal when the bracket is cinched down tight. Pitch pans around the roof penetrations may be required in some areas. A new tile can be made from cement mixes, using adjacent tiles as a form (and tinted, if required). Aluminum and copper materials should be protected from contact with the cement by a layer of tar to prevent corrosion.

Fastening schemes have been proposed which rely upon sealing the roof penetration at the tile surface. Since the waterproof membrane is the slate material beneath, these methods are not effective and should be rejected. Insist on a tight seal between the bracket and the membrane itself.

Spanish or barrel tile roofs present particularly tough collector installation problems. It is extremely difficult to work on them without breaking a significant number of tiles, and it is also difficult to make new replacement tiles and to seal penetration points. These jobs will reflect unavoidable additional labor costs.

Piping to Collectors

For low-temperature collectors, plastic pipe can be used in the plumbing from the pool pump to the solar collectors. PVC and ABS pipe (Schedule 40) are the most commonly used materials for this particular application and have performed satisfactorily. Neither material can withstand high temperatures. Due to the moderate operating temperatures, pipe insulation is not required.

Medium-temperature collectors may require copper plumbing. In normal operation, average flow rates through typical medium-temperature collectors are usually less than those for low temperature collectors; thus the return water will be warmer (although there will be less of it). PVC and ABS can accommodate this return temperature, which is generally below 100° F. However, when this type of collector stagnates, it may reach 200° F or more. Any water left standing in the collectors, and that which flows through them initially when the system is turned on, can become hot enough to damage plastic pipe. Consequently, copper pipe is used until the return water is diluted and cooled by the flow bypassing the solar system.

Further, water left in collectors when stagnation temperatures exceed 212° F will boil, forcing heated water or steam into the system plumbing. This situation may occur whether the filtration pump is on or off. Vacuum relief valves can vent some of the troublesome steam, and plumbing which facilitates complete drainage of the collectors when the system is shut off can minimize the problem. Isolation valves can be strategically placed to isolate the collectors from parts of the system that may be plumbed in plastic (to provide protection during the summer season).

If copper pipe is used to plumb medium-temperature collectors, it is usually covered with 1/4- to 3/8-inch foam insulation which, if exposed to the sun, must be protected from ultraviolet degradation. UV-resistant vinyl paints or wrapping tapes are available to protect the foam.

Standard practice should be observed when installing piping leading to the collectors. Since large-diameter pipes are quite heavy when filled with water, sturdy supports will be required. Pipe cuts should be deburred before assembly to prevent stray particles from clogging the small fluid passageways. By using the correct cement for the job and cleaning fittings with solvent, leaks can be avoided. Finally, because plastic expands and contracts considerably as its temperature varies; allowances should be made for change in length.

Flow Control and Safety Devices

The most important component in the flow control system is the automatic control valve normally installed in the main filter flow path after the collector feed line and before the collector return. (See Figures 12-14 and 12-15 on the following page.) Although this valve can be as simple as a manually-operated gate valve, it is generally operated automatically for convenience as well as for overall system efficiency.

A variety of control valves are available, but approximately 80% of all installations use either a bladder-type pinch valve or a specially-constructed variation of an irrigation valve.

The pinch valve (Figure 12-14) is one of the earliest automatic valves used on solar swimming pool systems. It consists of a flexible rubber cylinder that accepts pipe at either end and has an inflatable rubber bladder inside, which can be pressurized to shut off the flow. Small plastic tubing through which the bladder is inflated and deflated is connected to a fitting on the valve and terminated at the control box.

The valve is governed by an electronic control which measures and compares collector temperature to pool temperature. The upper (collector) sensor is mounted in a dull plastic housing that has an absorptivity for solar energy

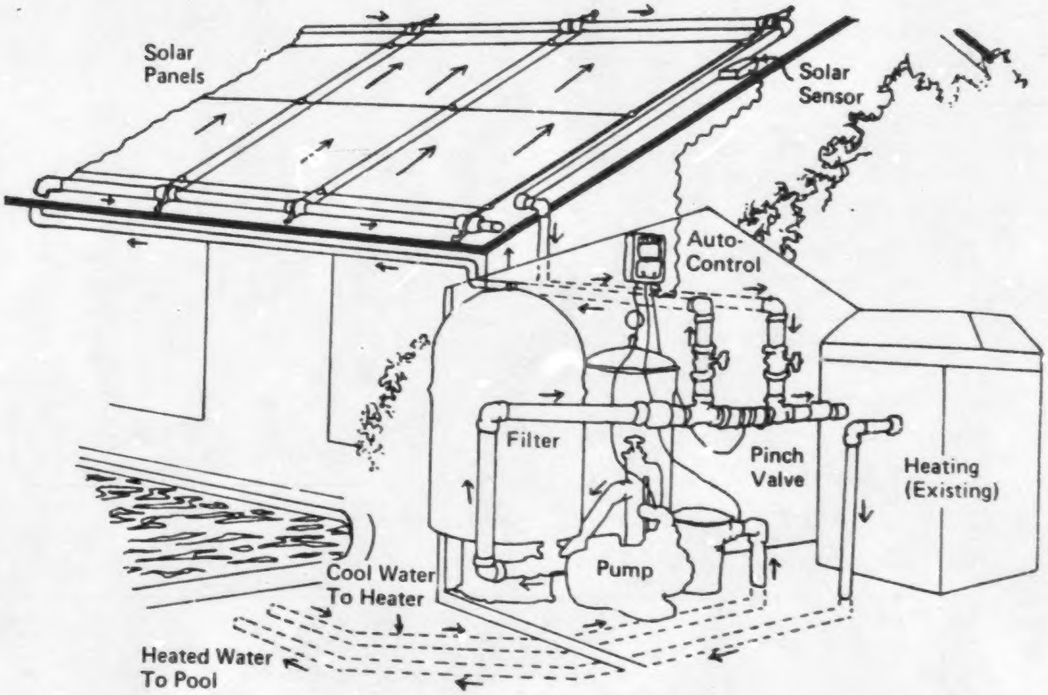

Solar Panels
Solar Sensor
Auto-Control
Filter
Pinch Valve
Heating (Existing)
Pump
Cool Water To Heater
Heated Water To Pool

Figure 12-14
Plumbing Schematic

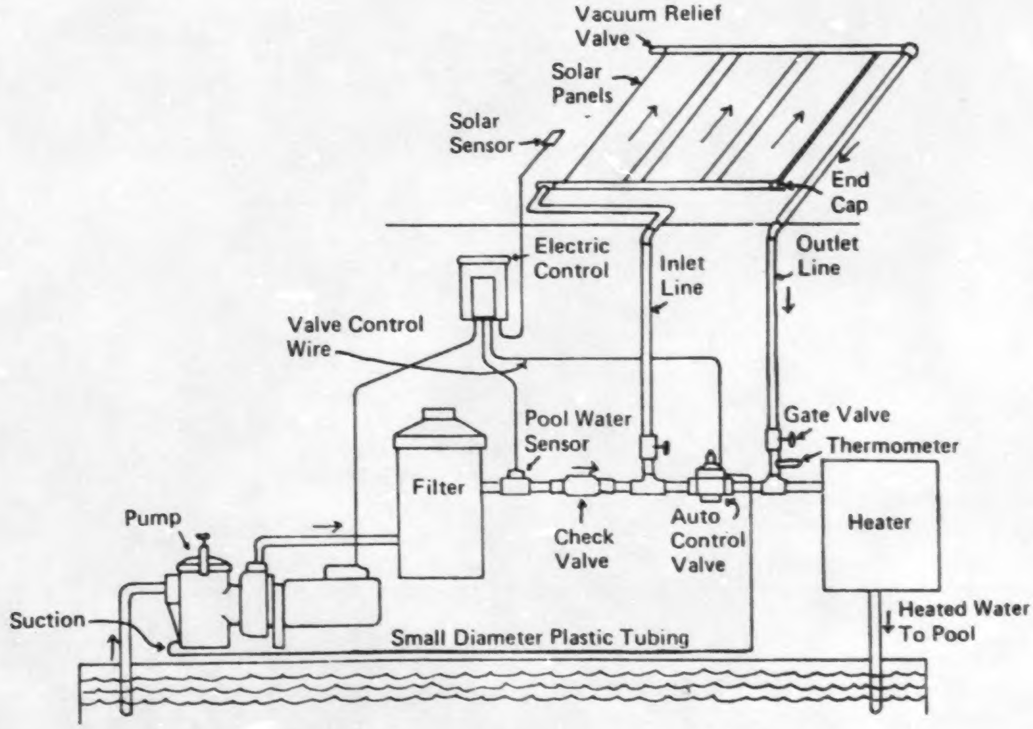

Vacuum Relief Valve
Solar Panels
Solar Sensor
End Cap
Electric Control
Inlet Line
Outlet Line
Valve Control Wire
Pool Water Sensor
Gate Valve
Thermometer
Filter
Check Valve
Auto Control Valve
Heater
Pump
Suction
Small Diameter Plastic Tubing
Heated Water To Pool

Figure 12-15
Plumbing Schematic

approximating that of the solar panel. Since the sensor temperature should simulate the collector temperature, it should be mounted alongside a collector panel and fastened to the same surface to which the panels are fastened. Wires to the sensor carry low-voltage electricity only; an electrician is not required for their connection. All wire connections should be made secure and watertight, preferable with heat-shrinkable insulating tubing or vinyl electrical tape.

The lower sensor, which measures pool temperature, should be immersed in the flow before the feed line going to the collector. It may be installed in a reducing-T, or a hole may be drilled directly into the pipe and tapped with a pipe tap. A small amount of teflon tape on the threads is recommended. If possible, the sensor should be installed in the bottom of the pipe to protect it from the sun's rays, and it should be heavily insulated to measure pool temperature rather than air temperature.

The electronic control itself requires electrical power. It can be wired to the pool pump or timer, but since 120V or 240V electricity is involved, you should consult your local building officials to determine if an electrician is required to make the connection. Approved conduit should be used for this wire.

The final step in the installation of this control system is the connection of two pressure lines between the pool pump and the control box (Figure 12-14). Small-diameter (1/8-inch) plastic lines are generally used. Since most pool pumps have a 1/4-inch, threaded pipe plug in the side of the strainer housing (near the bottom), the low-pressure line is usually attached there. The high-pressure line should be tapped into the pipe on the discharge side of the pump.

The control function is straightforward. When the upper sensor signals that the collectors are warmer than the pool water, the electronic control opens the line from the discharge side of the pump and inflates the bladder in the pinch valve. This diverts flow through the solar system. When the upper sensor signals that the collectors are cooler than the pool water, the line from the suction side of the pump is opened, forcefully deflating the bladder and allowing flow to bypass the collectors.

Rubber pinch valves have operated successfully for many years. A primary failure used to result from abrasion of the bladder by jagged pipe ends, but protective washers are now used to prevent that problem.

The second most popular control valve, the irrigation style valve, achieves the same results but operates in a slightly different fashion. Its squat, compact body is plumbed into the system using standard techniques. A small suction line is tapped into the pump inlet strainer housing, but in this case it is connected to the valve body itself. A pressure line is not used. A low voltage wire also connects a small solenoid valve mounted on the valve body to the control box.

The control compares collector temperature with pool temperature as before. If solar heat is available, an electrical signal alerts the solenoid valve to open the suction line. This suction closes the main valve, diverting flow through the collectors. When solar energy is not available, the solenoid valve remains closed, the main valve remains open, and the flow bypasses the collector array.

SIZING, DESIGN AND RETROFIT

INSTALLATION, MAINTENANCE AND OPERATIONAL CONSIDERATIONS

STUDENT MATERIAL

SIZING, DESIGN AND RETROFIT

INSTALLATION, MAINTENANCE AND OPERATIONAL CONSIDERATIONS

INSTALLATION AND OPERATIONAL CONSIDERATIONS

A. Arrangement of Components

1. Collector Orientation and Tilt

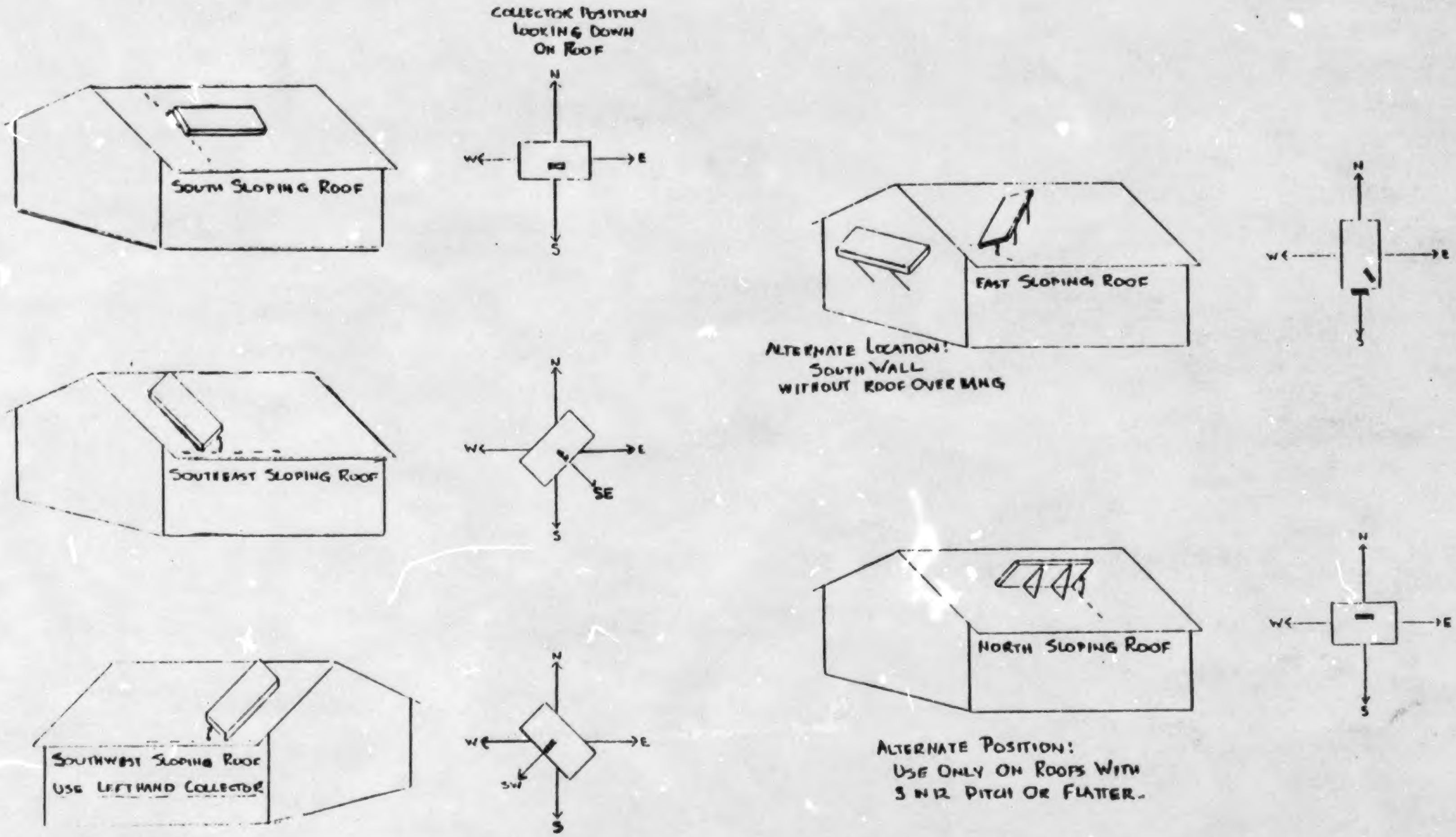
While solar collectors are often mounted on the roof of a building, they can be mounted on special racks on the ground. Select a suitable location, as close to the storage vessel as possible, with a southward orientation and proper tilt. (Remember: a "drain-back" system requires that the collectors be at a higher elevation than the storage.)

Try to avoid shading of the collectors by trees, chimneys, buildings, etc., keeping in mind that the sun is lowest in the sky in the winter and highest in the summer. It rises south of east and sets south of west in the winter, while in summer, it rises north of east and sets north of west. Most of the useful energy falling on a stationary collector is between 8 a.m. and 4 p.m. (solar time), so do not be overly concerned about early morning or late afternoon shading.

Solar collectors should normally be oriented toward the south. Due south is best, but for water-heating systems, deviations of up to 30° east or west of south cause only minor reductions in output (typically less than 5%). However, an orientation of 45° east or west of south will result in a noticeable reduction (typically 10% to 15%). For space-heating systems, the south orientation is somewhat more critical, because the system is used primarily in the winter, when the sun is lower in the southern sky. If the area has a typical morning or afternoon cloudiness, it is advisable to shift the collector orientation about 15° to the east or west to accommodate the sunny time of day. (See Figure 13-1 for different ways to achieve proper collector orientation on sloped roofs).

One way to check the true orientation of a collector that is mounted on an

Figure 13-1


COLLECTOR POSITION LOOKING DOWN ON ROOF
N
W
E
S
SOUTH SLOPING ROOF
SOUTHEAST SLOPING ROOF
SE
SOUTHWEST SLOPING ROOF
USE LEFTHAND COLLECTOR
SW
EAST SLOPING ROOF
ALTERNATE LOCATION: SOUTH WALL WITHOUT ROOF OVERHANG
NORTH SLOPING ROOF
ALTERNATE POSITION: USE ONLY ON ROOFS WITH 3 IN 12 PITCH OR FLATTER.

easterly (or westerly) sloping roof with one side tilted toward the south is to pour a small stream of water on the collector and mark its course as it runs down the collector face. This indicates the direction of the true orientation.

Do not guess directions. Always use a compass (or other accurate means), remembering that a compass points to magnetic north and must be corrected for true north. The sun rises due east and sets due west only on March 21 and September 21.

The collectors should be tilted up from horizontal by an amount equal to the local latitudes, plus 10° for water-heating, and 20° for space-heating. Check a Rand-McNally or other map for the local latitude.

2. Collector Arrangement

Collectors can be connected in series, in parallel, or in various combinations. For large groups of collectors, it is customary to have several collectors (5 to 15) connected in parallel using internal headers to form an array. Two or more arrays are often connected in series to form a bank, and several banks may be connected in parallel to complete the system (see Figure 13-2). As a general rule, parallel connected collectors have higher efficiencies that do series connected collectors.

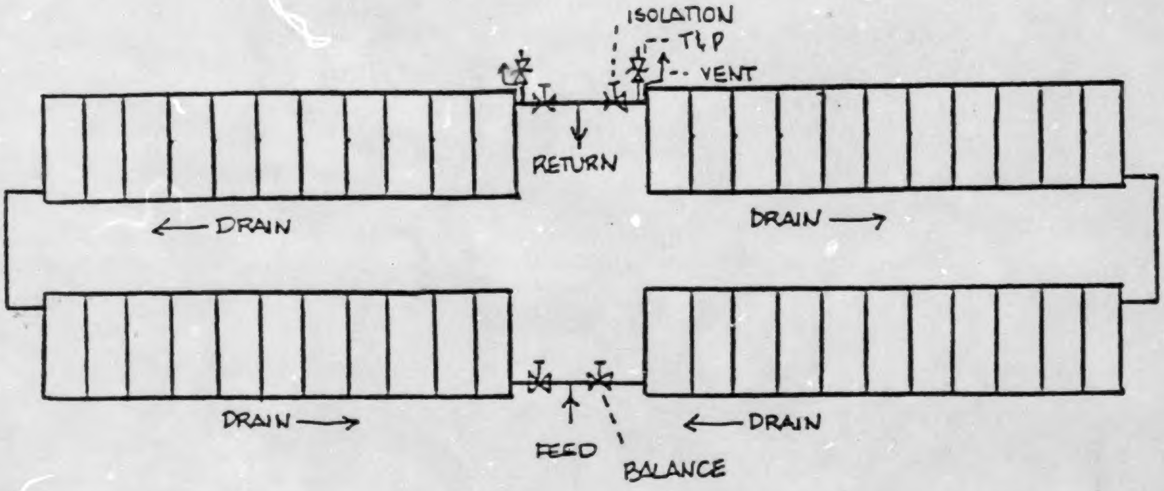


Figure 13-2: Typical Large Group Arrangement
Using Internal Manifolds

(a) With non-pressurized collector systems, the flow path must be constructed so that the system will completely fill. This can be accomplished by feeding in water at a low point and returning from a high point, forcing all the fluid to this high point before it returns to the tank (see Figures 13-3A and 13-3B)

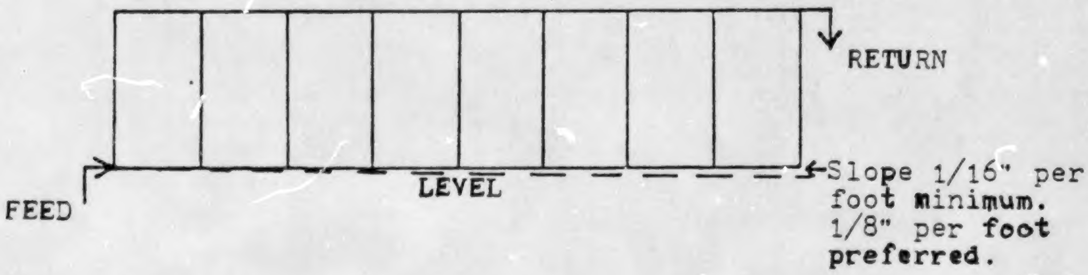


Fig. 13-3A: An Array of Collectors with Large Headers.

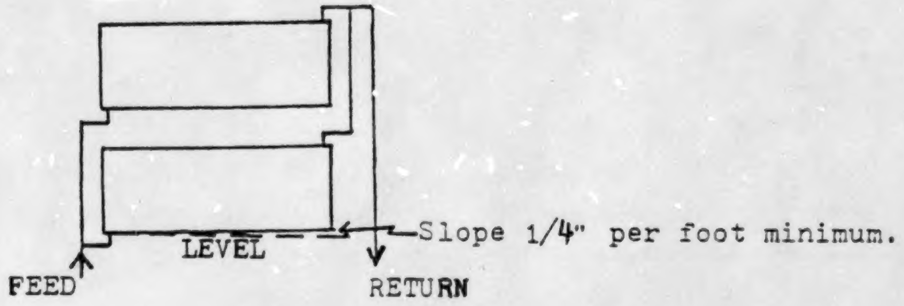


Fig. 13-3B: Individual Collectors with Horizontal Tubes

Note that if the collectors in Figure 13-3A were sloping in the opposite direction, the higher ones would not necessarily fill, and only the lower ones would produce any useful heat. Another common mistake in such configurations is to feed and return both from the left end. This compounds the filling and flow-balancing problems because of the additional dynamic head which develops in the collectors farthest away from the feed and return lines. In an arrangement such as Figure 13-3B, if the return from the lower collector were simply connected with a T into the return line from the upper collector, the upper collector generally would not fill.

(b) Again, it may seem unnecessary to say that a system designed to drain must be installed so that the collector and piping will drain, but the fact remains that this simple requirement is often overlooked by beginning installers. One-eighth inch-per-foot slope is normal for 1/2" and 3/4" pipe. Large pipe (1-1/2" diameter) will drain adequately with 1/16" per foot. Small pipes inside collectors should be sloped at least 1/4" per foot. (Never guess that a roof is level. Use a spirit level, and check collectors and piping). Soft copper pipe is convenient to use where there is plenty of slope, but where the slope is shallow, always use hard pipe. The soft pipe will sag between supports and not drain adequately. If there is any question about the long-term stability (resistance to sagging) of the roof or support, use a greater slope.

(c) When several collectors are connected in parallel, care must be taken to assure that all paths have approximately the same dynamic resistance and hence the same flow rate. Large collector fields with hundreds of collectors usually have balancing valves and flow indicators. For small and medium size systems, it is often possible and usually preferable to avoid the balancing valves, because of the trouble in establishing and maintaining a balanced condition. Install a temperature and pressure relief (T & P) valve for each section of pipe that can possibly be closed off by valves.

3. Storage Locations

The storage for a solar hot water system may assume various forms and locations. For a system with an electric auxiliary heater, it is customary to combine storage and auxiliary heating in a single tank, using thermal stratification to keep the coldest water on the bottom. This has the advantage that the tank can often be placed in the regular water heater closet. For many retrofit installations (and for all gas-type auxiliary systems), a separate storage tank is used. This storage tank can be placed next to the regular water heater, in a corner of the garage, or wherever space is available.

An existing house may have two or more water heaters, and the solar storage tank may be used to feed heated water to both (or several) of them. If possible, place the solar storage tank nearest the water heater that supplies the kitchen. Similarly, in a new house, it is best to locate the water heater nearer the kitchen and, if possible, near the center of hot water distribution. A typical hot water distribution system uses a 3/4" pipe that loops from one location to another. A more preferable method is to use several 1/2" lines, each running directly, like the spokes of a wheel, to bathrooms, utility room, etc. One-half inch pipe contains only one-half as many gallons of water per 100 feet of pipe as 3/4" pipe does, and if routed along a direct line, is shorter. This means that hot water is available at a remote location much faster and with less wasted energy and water.

All hot water lines should be insulated, even those embedded in the slab. Foam rubber insulation (1/2" thick wall) is easy to use and does a good job. Circulating loop-pumps or gravity circulation systems are convenient, but they are real energy wasters, and are to be avoided whenever possible.

There is a considerable advantage in placing the storage for a solar heating system inside the heated space, since losses from the storage tank are then not losses at all because the escaping heat is used to warm the house. The typical storage tank is a 500-to-800-gallon tank (3-1/2 ft. to 4-1/2 ft. in diameter and 6 ft. tall), so the space required is very modest. Of course, a solar heating system with an inside tank should be turned off in summer, so as not to add to the air-conditioning load. (Other locations, incidentally, are a small building behind the house, part of a tool shed, underneath part of the house built on a slope, etc.)

Underground burial of storage tanks is not generally recommended, though in some cases it is an acceptable method. Potential problems include: the possibility that tank insulation will be spoiled by ground water (some insulation is supposed to be water-proof, but there are questions as to its long-term ability to resist water completely); in many parts of the country, excavation is costly because of rocky ground; the servicing of pumps and/or heat exchangers can be awkward with buried tanks.

4. Pump Locations

Centrifugal pumps (the type most commonly used) should be located below the water level of the storage tank for all non-pressurized systems. Some pool pumps (once primed) will self-prime at an elevation of about three feet above water level, but these are more reliable if located below water level. The pump to the tank should enter the side of the tank, below water level, and should not have elevated sections that would prevent automatic priming of the pumps. (There are various schemes for avoiding openings into the side of a tank, but most of them are dubious and likely to lead to ruined pumps).

The pump in a closed-loop (anti-freeze) system is sometimes installed with isolation valves on each side, so that it can be removed for servicing without having to drain the system. (Some pump brands have isolation valves in the connecting flanges.) Shaft seals on pumps are a high-service item, while pumps with magnetic drive or fluid-encased motor rotors are less troublesome. When possible, pumps should be located where they can be easily serviced.

5. Vent Valve Location

A vent valve is used in a pressurized system to let trapped air escape from a section of pipe, usually from the collector circuit. Often placed at the highest point in the loop, it may be placed farther from the pump than the highest point, but never at an elevation that is more than H feet below the highest point (H being the feet of heat that the pump will develop at zero flow). The vent valve should be rated for the maximum pressure likely to be experienced by the collector system. (Many vent valves, rated at 15 to 25 psi, are unsuitable for use in connection with city water mains.) Some vent valves also act as vacuum relief valves, which is good, but others have check valves in the exhaust port to prevent re-entry of air; avoid these, unless a separate vacuum relief is provided. In an automatic drain-down system, a second vacuum relief valve should be placed where it will not ice over during inclement weather.

B. Combination Systems

Solar water heating and space heating are sometimes combined into a common system.

Advantages

1. Reduced stagnation problems for heating collectors during the summer.
2. A heat exchanger for the water heater, placed in the big tank, may cost less than a separate storage tank;
3. Supplies 100% of hot water needs during the summer months - a marginal advantage, because most independent water heating systems supply virtually all the hot water needs in summer anyway.

Disadvantages

1. The big pump motors consume power all summer, possibly costing more to operate than they save; yet, if the pumps are operated only two hours per day, the stagnation problem is aggravated;
2. The heating system will last twice as long if it is covered and shut down when not required (a 40-year life vs. a 20-year life);
3. The combined solar system is not suitable for inside storage, because the hot tank will lose heat to the house and increase the air-conditioning load.

A solar space-heating system is often used to heat a swimming pool in the spring and fall, when the heat is not required in the house. This is a good way to get added use from the solar investment, requiring only a little additional piping and some change-over valves (if it is a drain-back system).

A combination solar heating and solar cooling system has a lot of emotional appeal, because heating and cooling are both major energy users. The only solar cooling that is commercially available, however, is absorption cooling, and the cost of such equipment is almost prohibitive for home use. The prospects for long-term reductions in cost are discouraging, though larger absorption units for commercial use are somewhat more economical. Research work is underway on other types of solar cooling, such as desiccant cooling and solar driven turbine power, but their ultimate practicability is not known at this time.

C. Handling Stagnation

A solar collector exposed to the sun without benefit of a circulating coolant is said to be stagnating. The better a collector is insulated, the higher the stagnation temperature. A double-glazed flat-plate collector may stagnate at 400° F or higher on a sunny day. Since many collectors will deteriorate fairly repidly under these conditions, it is common practice to design collector systems that seldom, if ever, stagnate. This is usually easy to accomplish for water heating systems, because the equipment is typically used all year.

Space heating systems, however, are often idle during the hottest part of the year, which means that some method of avoiding stagnation must be found. The following methods are in general use:

1. Year-round operation: In this method, which is particularly applicable to combination heating and water heating systems, collectors are often tilted an extra 10° up from horizontal to minimize summer input, and the storage-collector fluid is pumped through the collectors at night for cooling as required to prevent excessive tank temperatures.

2. Covering collectors in summer: This method may be a problem (or a nuisance) for some homeowners, but it makes the equipment last longer and saves on pumping power.

3. Using collectors that are stagnation-resistant: Some collectors now available have reasonably good stagnation resistance, especially the single-glazed ones. In designs where there is likely to be considerable summer stagnation, such as enclosed swimming pools, space-heating systems, etc., single-glazed units with tempered glass covers are preferred. A black chrome selective surface absorber is recommended for space-heating systems, due to its improved winter performance. (Even with the most durable collectors, however, stagnation is to be avoided where practical).

D. Mounting the Collectors

The solar collectors must be securely attached to a roof or other stable structure. Wind-loading is usually a more important factor than the weight of the collector, and in northern parts, snow and ice loading are also important.

The action of wind as it blows across a sloped roof is difficult to determine accurately because the wind direction in relation to the house is a major factor. The force of the wind acting on the collector will normally be equal to or less than would on a vertical wall of a building. For most areas, 30 lbs per sq ft (100 mph wind) is considered adequate, though in coastal areas that are subject to hurricanes, you may wish to use a higher figure.

(Consult your local building code.) Multiply the surface area of the collector times the sine of the tilt angle to determine the net effective area. (A minimum wind load of 20 lbs per sq ft is recommended, regardless of the tilt angle, because large suction loads can be created even on flat roofs and flat collectors.)

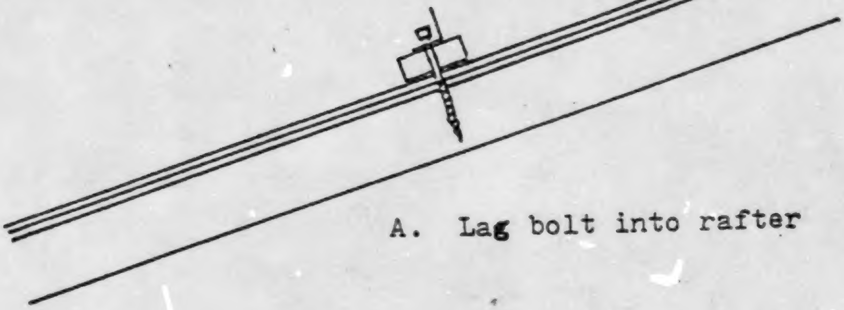
Example: For a 4' x 10' collector mounted at 45°, this would be: 4' x 10' x $\sin 45^\circ$ x 30 lbs/ft^2 = 849 lbs. If the collector extends above the peak of a sloped roof, then the full area of that portion of the collector should be used in computing effective area, rather than the sine of the tilt angle, because the wind coming over the ridge of the house may have an upward component with additional lifting force. (Wind-loading should be considered as both a positive and a negative force.) It is apparent from the sample calculation that the wind load is substantially more than the 100-to-200-lb weight of the typical 4' x 10' collector (filled with water).

Two inches of solid ice, or 20 inches of snow, weighs about 10 lbs per sq ft. Although a 50-to-60-mph wind might accompany an ice or snow storm, it would be very rare to have a 90-mph wind at the same time, so the wind loading and the ice loading are not additive. In snow country, blowing snow can pack under collectors and add to the snow load on the roof. This is not often a problem on the sloped roof of a house, but may be an added load if many collectors are put on a flat roof. (Note: because the collectors are warmer than the roof, it is common for snow to melt and run off the collectors, fall to the roof and freeze, causing an ice build-up at the base of the collectors. Thus, in snow country, it is often desirable to mount the collectors in such a way that these ice dams do not cause water back-up on the roof.)

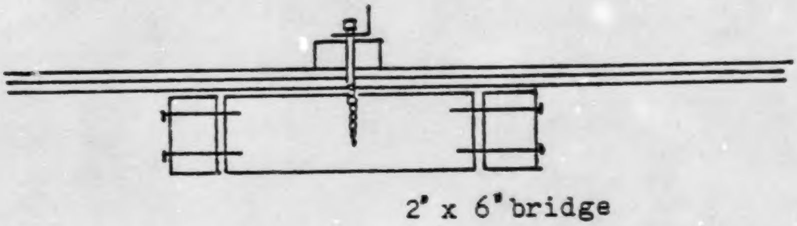
Based on the above description, it is obvious that the solar collectors must be securely mounted to withstand wind and other forces. While it is best to fasten the collectors directly into the rafters if possible, two alternative methods are shown in Figure 13-4, which is on the following page.

1. Sloped Roof Mounting

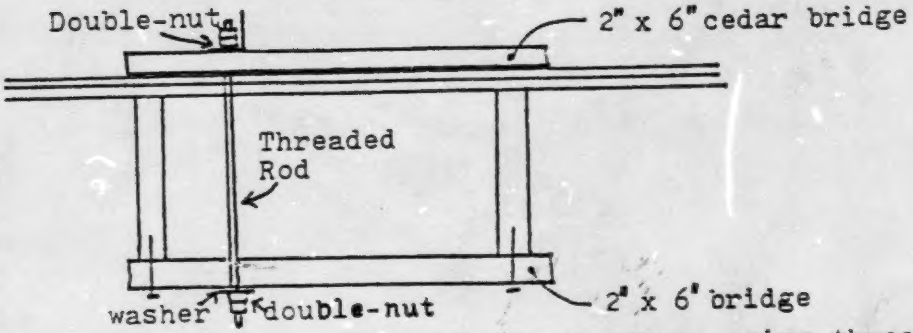
It is important to locate the center of the rafter so that the lag bolts will have full holding power. One way to do this is to drill a 1/8" hole from the



A. Lag bolt into rafter



B. Lag bolt into wood bridge between rafters.


Double-nut
2" x 6" cedar bridge
Threaded Rod
washer
double-nut
2" x 6' bridge

C. Wood bridge above and below rafters using threaded rod and nuts.

Figure 13-4

underside of the roof, starting at the edge of the rafter, which is slanted toward the center at such an angle (typically, 45°) that the drill will penetrate the top roof survace directly over the rafter center. Then a 1/4" hole can be drilled from the top side (perpendicular to the roof) through this small hole.

If the underside of the roof is not accessible, it may be necessary to drill some 1/8" test holes. With a composition shingle roof, a shingle can be heated and raised so that the test holes fall beneath a shingle. Fill the test holes with silicone caulk and screw in 1" #10 flat-head screws until they are flush.

Use 1" thick cedar or redwood blocks (4" x 4" x 1") at each bolt location to space the equipment off the roof. (Sometimes it is necessary to use thicker or additional blocks at one side of the collector to achieve a proper slope for draining.) Coat the blocks with silicone, fill the holes with silicone, place a ring of silicone sealer around the holes, and screw in the lag bolts. Since the lag bolts must penetrate 3" into the rafter, you will probably have to use 5" bolts or longer (Figure 13-4). (An alternate method of bridging is shown in Figure 13-4C, but since it required a bridge on top of the roof as well as on the bottom, it is not as desirable. Also note the alternate use of threaded rods and nuts instead of lag bolts.)

Where pipes penetrate the roof, use flashings to seal around the pipe. Drill a hole in the roof, and put a ring of silicone sealer on the bottom of the flashing. Place the top edge of the flashing under the shingle above the hole, screw the bottom edge of the flashing to the roof, and seal around the screws. After the pipes are run through the flashings, seal with silicone sealer and pull the pipe insulation down over the top of the flashing.

2. Flat Roof Mounting

All roof penetrations through a flat roof should be sealed with pitch pans (a small pan or box that will hold about a quart of molten tar, which is poured around the pipe or mounting bracket). Scrape away any gravel or rocks with the claws of a claw hammer (in warm weather, scrape away the gravel early in the morning, when the roof is cool). Drill holes, run pipes through the roof, and attach mounting brackets. Place pitch pans over the pipes and mounting brackets, and cement in place with silicone sealant. Stuff the crack between the hole and the pipes with a piece of rag. Melt roofing tar and pour into the pitch pans (see Figure 13-5).

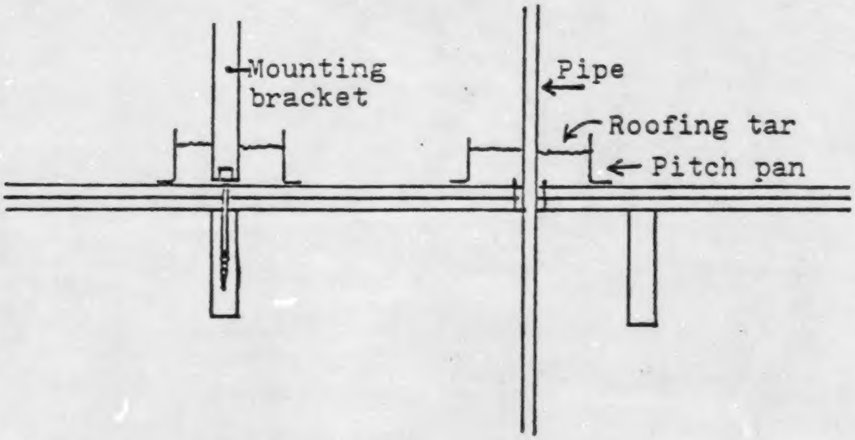


Figure 13-5

Large arrays of collectors on commercial buildings may occasionally require a separate steel structure attached to the steel framework of the building. An analysis should be made by a qualified structural engineer to determine whether the roof structure is adequate for the additional loading.

3. Tile Roof Mounting

Tile roofs require a special collector mounting. (Because such roofs are fragile, moreover, the installer must work and walk very carefully.) The mounting brackets must be supported in such a way as to avoid putting weight on the tiles. One way to do this is to drill a hole through the tile with a 1-1/2" masonry core drill. The collector mounting bracket can then be attached to the rafter, using a pipe-spacer and a long lag bolt (see Figure 13-6, on the following page).

After putting silicone sealant in and around the hole in the rafter, place a washer between the pipe and the roof sheeting. Then seal the washer and the pipe with silicone sealant, as well as the space between the pipe and the tile, building up a slight mound of sealant around the pipe.

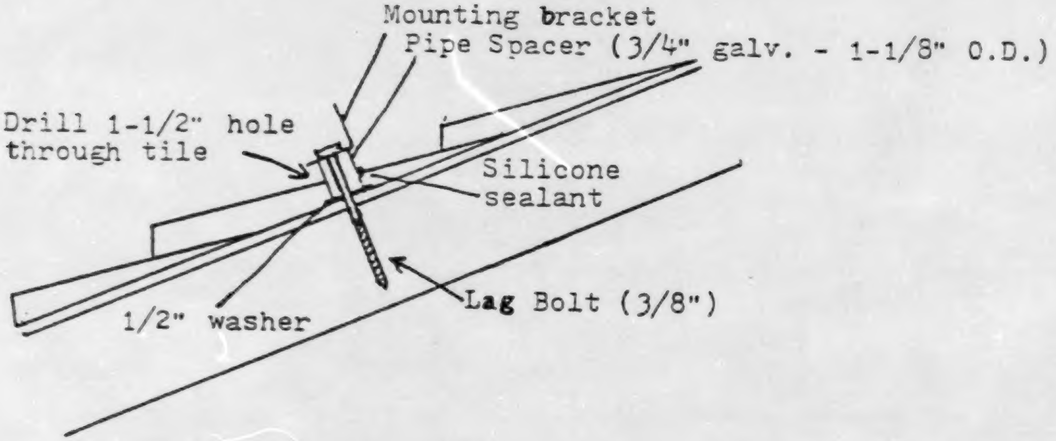

Mounting bracket
Pipe Spacer (3/4" galv. - 1-1/8" O.D.)
Drill 1-1/2" hole through tile
Silicone sealant
Lag Bolt (3/8")
1/2" washer

Figure 13-6.

4. Mounting Thermosipyon Systems

As with all systems, a location near the regular water heater is desirable, because this means shorter pipes and reduced heat loss. Select a south-facing roof at least 14' wide (if possible), and try to slope the collector due south (though the slope direction can be anywhere from 45 degrees east of south to 45 degrees west of south, without requiring special mounting).

The tank needs to be mounted with its upper edge at the peak of the roof. The roof is stronger at the peak than farther down the roof, and the tank should be as high as possible to enhance water circulation. In fact, the bottom of the tank must be at least one foot (measured vertically) higher than the top of the collector. This means that, on a normal sloped roof, some 3' to 4' of separation between collector and tank will be required. In a typical configuration, about 14' of roof is needed (8' collector and 4' separation, plus 2' tank, equals 14' of sloping roof). If the roof is not wide enough, the collector may be turned sideways or the tank raised on timbers, or both as required. Fasten the tank and collector to the roof with 3/8" (diameter) lag bolts sunk into the rafters.

SCHEDULING INSTALLATIONS

There are several sequential and concurrent activities in operation when installing a solar system during new home construction. Scheduling is important.

With a carefully established plan of attack, the builder will have fewer unexpected delays during construction. Delays cause increases in building construction costs.

Gathering all the materials necessary to complete an installation is a preface to scheduling the job. One survey noted that nearly 1/3 of the problems encountered by builders of solar systems were delays in delivery of major components.

The type of solar system to be installed affects scheduling. Many areas of construction are the same whether the system is liquid or air. However, there are a few major construction areas that differ. For example, there are differences between scheduling water storage installation for a liquid system, versus rock storage for an air system. The type of building to have the solar system could make a difference in some of the scheduling. A residential installation could differ from a commercial installation at various points in the scheduling.

Location of the project, seasons and weather are other factors that could affect the initial schedule and/or cause schedule adjustments after the job is started. Experience will be a great teacher when scheduling solar installations. At present, the solar industry is too new to make unbending judgments on some scheduling problems.

It is recommended that the installer construct a flow chart or PERT chart for installation milestones that must be used, if the job is to be completed in a timely manner. (See Lennox Solar Training Manual for example).

GENERAL MAINTENANCE AND PERFORMANCE PROCEDURES

VISUAL INSPECTION

The first step in a practical check-out procedure is a visual comparison of the completed system with construction drawings and manufacturers' instructions. Fluid flow patters, sizes and locations of ducts and piping, and sealing of duct joints should be verified. Electrical connections on controls, sensors, and motors should be checked for integrity and conformity with plans.

The collector installation should be inspected for alignment, tight connections, properly positioned piping and ductwork, and security of collector support and perimeter flashing.

After completion of the visual inspection of the idle system, various observations should be made when the equipment is operated in each required mode. Generally, the following checks will usually be necessary:

1. Observation of leakage of liquids or air at any point in the system, while running in each operating mode;
2. Verification of proper functioning of all moving parts and controls, including motors, pumps, fans, motorized valves and dampers, manual valves and dampers, check valves and dampers, fuel valves, electrical switches and relays, thermostats and their contacts, draining and venting valves and fittings, and any other adjustable components in the system;
3. Liquid levels and compositions;
4. Cleanliness and freedom from obstructions in flow channels for liquids and air;
5. After initial operation, cleanliness of filters in liquid and air circuits;
6. Verification of proper control settings and their proper action (turning on, turning off) in actuating motors, valves, and dampers;
7. Detailed check of proper functioning of safety and limiting devices and designs, such as complete drain-down of water from collectors and piping in drain-down systems, automatic valves for draining, venting, and bleeding liquid systems, overheat protection devices such as pressure relief valves and sensor actuated drain valves, and any other system protection devices.
8. Complete check of auxiliary heating and distribution system.

The check-out procedure outlined above should be followed prior to final insulation of the principal system components and piping unless insulation is part of the equipment such as in internally insulated air ducts. The heat storage unit may, however, be insulated prior to final system check.

FINAL CHECK

Following the testing and the repair and correction of faults which may be found, insulation should be applied to components and interconnections in the system, as recommended or required. A final check on the proper functioning of all moving parts and control functions should then be made to eliminate possibility of malfunction accidentally caused by damage during the insulation process, and as a double check on system functions. This final check-out may advantageously be done in collaboration with the system owner, so that he may understand its operation.

OPERATIONAL CHECK-OUT

Of particular importance is the careful use of manufacturer's and designer's check-out instructions in verifying and correcting the installation and operation of each system and its components. Procedures are provided by some suppliers of complete solar heating systems, and the manufacturers of the principal components usually supply check-out information on their own products. Collectors, pumps, motorized valves and dampers, and control subsystems are examples

In checking operations in various modes, it is usually necessary for the system installer to simulate one or more conditions not prevailing at the time of testing. If the sun is obscured, for example, solar collector checking required imposing some type of artificial condition to actuate the appropriate components. The supplier of the controller or of the complete system usually offers instructions on simulating each operating mode by making jumper connections across sensor terminals. Sensor control settings must be verified, however, by observations under actual conditions.

PERFORMANCE TEST

The third major checking procedure is the measurement of heat delivery from the solar heating system. Although only a few systems are provided with convenient and accurate facilities for solar system efficiency checking, the installer can usually make a reasonable reliable determination of performance.

By comparing the measured results with design values, the quality of the system and its installation can be verified. As indicated in more detail below, measurement of inlet and outlet collector temperatures, and measurement or estimation (based on measured pressures) of collector fluid flow rate near noon on a sunny day, will provide sufficient information for at least a minimum evaluation of system performance.

A general blank form for use in checking a complete solar heating system is presented in the Appendix to this module. It includes numerous items which might be considered trivial or obvious, but even a minor fault, if overlooked and uncorrected, can cause poor performance, system deterioration, or building damage. Heating practitioners may also notice omissions which, in their experience, should be covered in the check-out process. Since the listing is intended to apply to all types of solar heating systems, some items are not applicable in each case considered.

Although sometimes difficult, measurement of solar heat collection and delivery to use provides the best assurance of good system performance. Fluid temperature rise and flow rate through the collectors, ambient temperature, and solar radiation should be measured. Solar heat collection and collection efficiency can then be calculated. If flow rates cannot be directly measured, they may sometimes be approximated by determining pressure drops across manufactured standard equipment, such as a heat exchanger or collector, in which the pressure-flow relationships are known.

AIR SYSTEMS CHECK POINTS

Among the check-out steps listed in the general procedure, several are particularly important in air systems. Air leakage, damper closure, blower and motor operation, and proper control should be thoroughly checked. Because of their specialized aspects, collectors, storage units, air handlers, and controllers require more than routine attention

The collector inspection check-list used by installers of a nationally distributed solar air heating system follows:

INSPECTION CHECK LIST - COLLECTORS

1. Collector Array: Refer to plans.
2. Holding proper dimension from collector to collector - thus making airtight seal from port to port: Refer to specifications and/or plans.
3. Used specified material for port and end cap sealant: Refer to specifications and/or plans.
4. Relief tubes sealed and in place: Refer to specifications.
5. Confirm location and dimensions of 2 x 8 (1-1/2" x 7-1/8") frame.
6. Cap strips installed so proper and airtight seal is accomplished.
7. Perimeter insulation installed: Refer to specifications and/or plans.
8. Perimeter flashings installed properly: Refer to specifications and/or plans.
9. Connecting Collars: Refer to plans.
    a. Location
    b. Sealed properly
10. Heat Sensors: Refer to plans.
    a. Installed properly
    b. Correct location

An inspection list for the pebble-bed heat storage unit specified by the same solar air system manufacturer follows:

INSPECTION CHECK LIST - PEBBLE-BED HEAT STORAGE UNIT

1. Location: Refer to plans.
2. Location of Duct Openings: Refer to plans.

3. Dimensions of Unit: Refer to plans.

4. Dimensions of Duct Openings: Refer to plans.

5. General Construction: Refer to plans.

   a. If construction does not follow plans, make sure modifications are adequate to meet specifications.
   b. Check for exposed wood - all combustible surfaces must be covered with a non-combustible material.

6. Check all joints:

   a. Sealed adequately
   b. Correct sealant used (i.e., suitable for temperatures around 180° F).

7. Lower Plenum: Refer to plans.

   a. Proper materials used
   b. Correct dimensions
   c. Correct spacing of bond beam block

8. Upper Plenum: Refer to plans.
   Correct Dimensions

9. Rock: Refer to specifications.

   a. Proper size
   b. Free of foreign materials (clean)
   c. Proper amount
   d. No depressions in rock bed (level on top)

10. Storage Unit Lid: Refer to specifications.

   a. Construction
   b. Sealed adequately
   c. Proper sealant used (i.e., suitable for temperatures around 180° F).

An air handler start-up procedure, which is effectively an installation check list, is shown below. This procedure also verifies the proper functioning of nearly all the elements in the control system.

START-UP PROCEDURES - AIR HANDLER

I. Preliminary Check

1. Double check all line and low voltage wiring and connections (see wiring diagram for exact wiring hook-up to unit).

2. Check all damper positions, both inside air handler and any dampers that might be located elsewhere in the duct system.

3. Check belt-driven power train (tighten set screws on pulleys, confirm V-belt alignment, etc.).

4. Check voltage supply to unit. Should voltage not be correct, contact electrician before proceeding with start-up.

5. Open all registers, diffusers and grilles in distribution system.

II. Start-up on "Sunny Days"

1. Set heat anticipators in space thermostat.

   a. W_1 (first stage heating) set at 0.7 amp.
   B. W_2 (second stage heating) set at 0.1 amp.

2. Set thermostat so W_1 is calling for heat. W_2 must not be calling for heat at this time.

   NOTE: The use of a jumper between W_1 and R_H at the control panel can be used in lieu of setting the thermostat. Before the system is given approval, however, check system operation with the thermostat to insure proper operation.

3. Set Sub-base switches (if present).

   a. "Fan-Auto"
   b. "System-Heat"

4. Turn on circuit breakers.

5. Turn on disconnect feeding the air handler.

6. Observe unit operation.

   a. The auxiliary furnace blower should be running. There should be no auxiliary heat.
   b. The air handler blower should start as long as sensors T_{co} and T_{ci} have a 45° F or greater temperature differential.
   c. Air flow through dampers in the air handler should be as shown on your plans.
   d. A temperature differential of less than 45° F will automatically switch the system into a "heat from storage" mode. If there is no heat in storage (less than 90° F), the control board will automatically by-pass the solar heating circuit and bring on the auxiliary heat source without having W_2 in the space thermostat in a "heat" position (closed circuit).
   e. When the rock storage unit has enough heat (greater than 90° F) available, and the T_{co}, T_{ci} differential is less than 45° F, the air handler will direct the air flow through the rock storate unit and into the auxiliary furnace. This will be accomplished without the auxiliary heat coming on.

7. Set the space thermostat so W_1 and W_2 are not calling for heat (open).

   a. The auxiliary furnace blower will cease operating
   b. The collector air handler will continue to operate
   c. The dampers inside the air handler will direct the solar heated air to the rock storage unit.

8. Set the space thermostat so W_1 and W_2 are calling for heat (closed circuit).

   a. The auxiliary furnace should operate in a conventional manner.
   b. The air handler blower will continue to operate.
   c. Dampers in the unit will direct air through the rock storage and into the auxiliary furnace.

PERFORMANCE CHECK

Knowledge of the heat output of a solar air heating system is of value to the owner. If measurement of air flow rate is impossible or impractical, static pressure readings at several places in the system can be used to obtain a reasonably close estimate of flow rate by comparing measured pressure drop across collectors, air handler, or hot water coil against manufacturer's data. Small holes (1/4-inch diameter) may be drilled or punched through duct walls at the positions shown in the Solar Heating Flow Schematic shown on the next page. Flexible tubing is then inserted for connection to a manometer or sensitive differential pressure gauge. Temperature sensors (glass or dial thermometers, thermocouples, or thermistors) are also inserted through holes at these points.

A performance analysis form used air system, with sample data, is shown in the following pages. Points for measurement of temperature and static pressure are indicated. By comparison of static pressure differences across the several components with those shown in the manufacturer's engineering data sheets at various flow rates, actual flow rates may be closely estimated. Heat output may then be calculated by completing the form entitled Performance Analysis, Air Heating System. By use of a simple hand-held meter, solar radiation can also be measured so that collection efficiency, item B2 in the Performance Analysis form, may be calculated. Comparison with manufacturer's performance data may then be made. If a large difference is found, causes should then be investigated.

LIQUID SYSTEMS

The principal components of a typical liquid space heating system requiring check-out procedures specific to the liquid type (as compared with air) are:

1. liquid pumps
2. liquid-to-liquid heat exchanger
3. valves and piping for draining and venting.

PROJECT NAME: Typical Project DATE: ______

PROJECT TYPE: RES'L X COMM'L ____ IND'L ____

AGRI ____ OTHER ____________

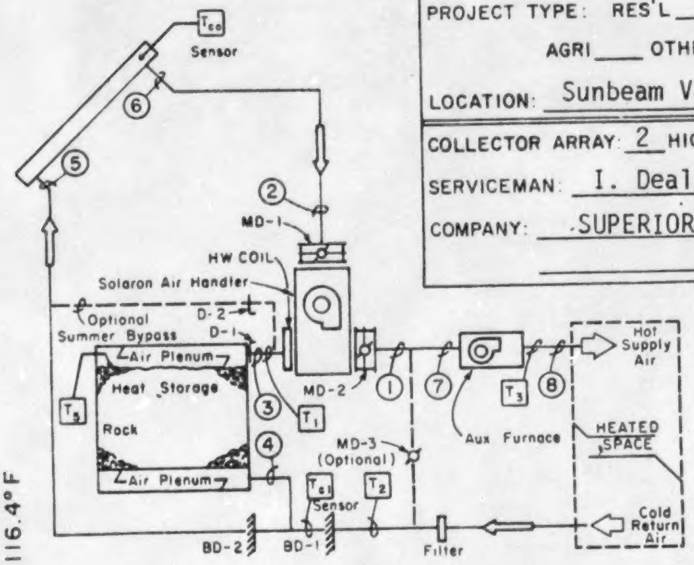
LOCATION: Sunbeam Valley, California

COLLECTOR ARRAY: 2 HIGH × 12 WIDE = 468 SQ ft

SERVICEMAN: I. Deal Heavenly

COMPANY: SUPERIOR SUNBEAMS, INC.

PHONE ( )



Tco
Sensor
6
5
2
MD-1
HW COIL
Solaron Air Handler
Optional Summer Bypass
D-2
D-1
Air Plenum
Heat Storage
Rock
Air Plenum
T5
3
T1
4
MD-2
1
7
T3
8
Aux Furnace
Hot Supply Air
HEATED SPACE
MD-3 (Optional)
Tc1 Sensor
T2
BD-2
BD-1
Filter
Cold Return Air

SOLAR HEATING FLOW SCHEMATIC

	AIR HANDLER	
	SOLAR	AUX
Design CFM	936	
Design Ext. SP	0.80"	
Fan RPM	1160	
HP	1/2	
Motor RPM	1725	
Volt	115	
Phase	1	
FLA	6.0	
SF	1.25	
SFA	6.8	
Insul. Class	B	
Motor Mfg	GE	
Model No	MT0051	

SEQUENCE OF OPERATIONS

O = Open
PO = Partially Open
C = Closed

	MD-1	MD-2	BD-1	BD-2	MD-3	D-4 / D-2	Pump	Solaron air handler	Aux Furnace Fan	Aux Furnace Heater
HEATING FROM COLLECTOR	O	O	O	O	P.O.	O / C	ON	ON	ON	OFF
HEATING FROM STORAGE	C	O	O	C	P.O.	O / C	OFF	OFF	ON	OFF
STORING HEAT	O	C	C	O	O	O / C	ON	ON	OFF	OFF
HEATING WITH AUX. FURNACE	C	O	O	C	P.O.	O / C	OFF	OFF	ON	ON
WATER HEATING (SUMMER)	O	C	C	O	O	C / O	ON	ON	OFF	OFF

FOR HEAT PUMP SYSTEMS: MD-2 Closed, MD-3 Open, BD-1 Closed

$$T_3 = \frac{T_1 (CFM_{sol}) + T_2 (CFM_{aux.} - CFM_{sol})}{CFM_{aux.}}$$

$T_1 = 130°F$
$T_2 = 68°F$
$T_3 = 117°F$

$$T_3 = \frac{130 (936) + 68 (1200 - 936)}{1200} = 116.4°F$$

TEMPERATURE & STATIC PRESSURE MEASUREMENTS

STORING HEAT		Motor Amps. 5.0		HEATING FROM STORAGE				HEATING FROM COLLECTOR		Motor Amps 5.3	
POINT	°F	STATIC PRESSURE	S.P. DIFF.	POINT	°F	STATIC PRESSURE	S.P. DIFF.	POINT	°F	STATIC PRESSURE	S.P. DIFF.
1		0.00"		1	-	0.00"		1	-	-.19"	.42"
2	138	-.34"	.69	2	-	-.01"		2	-	-.94"	
3	132	+.35"	.19	3	130	-.29"	.14	3	-	-.20"	.04"
4	68	+.16"		4	68	-.15"		4	-	-.16"	
5	66	+.06"	.31	5	-	-.01"	.00	5	66	-.34"	.42"
6	141	-.25"		6	-	-.01"		6	122	-.76"	
7				7	-	-.42"	.60	7	-	-.21"	.48"
8				8	117	+.18"		8	-	+.27"	

PERFORMANCE ANALYSIS

Air Heating System

A. Collector

1. Area, A_c ____________ ft^2

2. Inlet temperature, T_i ____________ ^oF

3. Outlet temperature, T_o ____________ ^oF

4. Pressure drop across blower, Wp ____________ in W. G.

5. Flow rate, $\dot{V}$ ____________ cfm

6. Air density, ρ ____.07____ lb/ft^3

7. Specific heat, c_p ____.24____ Btu/(lb· ^oF)

8. Heat delivery rate:

$$Q_c = (\dot{m}c_p)(T_o - T_i)(60)$$

$$= \dot{V} \cdot \rho c_p (T_o - T_i)(60)$$ ____________ Btu/hr

B. Collector Efficiency:

1. Solar radiation on horizontal surface, I_H ____________ Btu/(ft^2·hr)

   or on tilted surface, I_T ____________ Btu/(ft^2·hr)

2. Collector efficiency

$$h_c = \frac{Q_c}{I_T A_c} \times 100$$ ____________%

The other system components such as collectors, storage tank, motorized valves, pipe connections, sensors and controllers, all have their counterparts in the air systems previously described. If the supplier of a main component, such as the collector, has provided a complete system design and check-out procedure, those instructions should be followed. In other instances, particularly if large or custom-designed systems are involved, the mechanical engineering designer or consultant may specify a check-out procedure. If no specific inspection list is available, the installer should follow the general guidelines previously outlined, altering them as may be necessary for the specific system involved.

PUMPS

With respect to pumps, the installer should check speed (unless directly coupled to motor), and by means of permanent or temporary gauges, the pressure difference from inlet to outlet of the pump. Proper mounting, alignment, and attachment to piping and wiring should also be checked. Noise and vibration should be within acceptable limits.

By use of the pump speed and the fluid pressure difference, the pump manufacturer's charts or graphs may be used to determine volumetric flow rate. This rate should then be compared with the desired or designed flow rate and, if not in satisfactory agreement, causes for difference should be determined and corrected.

Because misleading pressure readings may sometimes be obtained at points near pumps, measurements of pressure differences across other components in the system should also be made. Pressure loss through the collector array and across a heat exchanger may be compared with the manufacturer's flow rate-pressure drop data to confirm the flow measurements. If serious discrepancies appear, their cause should be determined.

HEAT EXCHANGER

In dual-liquid systems, proper operation of the collector-storage heat exchanger must be verified. Counter flow of the two liquids is essential, so piping connections must be checked to verify the flow of one liquid in a direction opposite to the flow of the other. When operating under conditions such that solar energy is being collected, the inlet and outlet temperature of the collector fluid, the inlet temperature of the storage fluid, and the static pressure at all four points should be measured. If thermometer wells are not provided in the piping, temperature sensors (thermocouples, thermistors or small-bulb thermometers) should be tightly taped to the outside of the pipe in close proximity to the heat exchanger connections. At least 2" of pipe insulation should be applied over the temperature sensor along at least 1' of pipe length. After they become constant (several minutes) the readings will be sufficiently close to the liquid temperatures at those points. Calculation of a heat balance on the exchanger and a comparison of pressure loss with those shown in the manufacturer's data should then be made.

ANTIFREEZE SOLUTION

The concentration of antifreeze solution in the collector liquid should be verified by use of a hydrometer such as commonly available for testing automobile radiator coolants. If the test shows inadequate freeze protection, addition of ethylene glycol or propylene glycol, as specified by the designer, whould be made until the concentration is satisfactory. Tables of antifreeze concentration and liquid density may be used if the hydrometer does not show the freezing temperatures directly. Drainage of some liquid from the collector loop may be necessary in order that antifreeze can be added.

LIQUID LEVELS

The liquid level both in the collector loop and in the storage tank should be determined by whatever means are provided in the system. Sight gauges, pointer type indicators, inspection ports, and overflow valves or other means may be used, depending on design.

In a closed collector loop, proper filling and functioning of the expansion tank must be verified, sufficient liquid being present to fill the collector loop and sufficient additional space being available for the enlarged liquid volume when solar heated. Consideration must be given to the temperature of the liquid at which the expansion tank level is noted so that either an increase or decrease in liquid volume can be accommodated.

If a non-aqueous liquid is used in the collector loop, such as a silicone oil, its quantity and quality should be checked. Normal procedure would involve filling of the system by the installer, from a known supply of the chemical. In case of doubt, some chemical or physical property of the liquid which the manufacturer recommends for identification should be ascertained.

CORROSION INHIBITOR

If a corrosion inhibitor is used in the storage liquid, the most satisfactory procedure for insuring protection is careful compliance with designer's instructions when the inhibitor is first added to the system. If later verification of the quantity and quality of the additive is required, a chemical or physical test of a sample of the solution should be made by the installer. In some cases, a sample may have to be supplied to a testing laboratory for checking. In exceptional cases, the most economical precedure may be the draining and recharging of the storage unit, with properly measured additives.

DRAIN-DOWN SYSTEMS

Collector Loop

In drain-down systems, careful inspection of all piping and connections must be made to verify the absence of any low points or traps in the system that might prevent complete drainage. Even horizontal runs of pipe should be avoided, slight slope being a much preferred disign. Access of air to the collector either through an atmospheric valve or from the storage tank via an adequately sized pipe must be verified.

Systems that are designed for collector drainage to occur either when freezing threatens, or when the pump ceases operation, must be thoroughly and completely checked by dependable and repeated drainage and refilling, without fault. Those that are designed to drain back into the storage tank, either through an open return or siphon return when the pump ceases operation, can be checked by interrupting the power supply to the pump motor. After a minute or so, absence of water outflow through an opened drain valve located in the collector supply piping above the level of water in the storage tank indicates satisfactory collector drainage. The lack of water in the return line from the collector to the storage tank can be similarly verified. In siphon return systems, the opening of the "siphon breaker" air inlet valve at the top of the collector piping assembly must also be verified. This valve must be open whenever electric power is accidentally or purposely interrupted.

Start-up of the system after drainage must also be checked by restoring power to the pump and to the siphon breaker valve (if the valve is of the electric-operated type). Observation of flow returning to storage, if in a visible location, or verification of normal pressures at various points in the circulating loop can confirm satisfactory displacement of air from the collector and piping, and the restoration of normal flow. In the siphon return system, verification of proper functioning of the air bleed valve at the top of the piping array must also be made.

Storage Tank

If main storage is operated at a pressure greater than atmospheric pressure and is non-vented (usually because a pressurized hydronic heat distribution system is involved), proper operation of the pressure relief valve (safety valve) on the storage tank must be established. If the valve cannot be tested satisfactorily in place, it should be removed temporarily and tested with measured pressure by use of a pressure test kit. The greater complexity of the piping and valving in this type of system requires verification of complete collector drainage by methods recommended by the designer of the system employed. There is sufficient variation in the designs of these systems, including liquid level control, automatic water make up, check valves, bypass piping, and motorized valves, that there is no standard check-out procedure applicable to all systems.

Sensors and Controls

Systems which involve collector drainage only when freezing threatens are are usually actuated by a temperature sensor in the collector or in the atmosphere. These controls must be checked by artificially cooling the sensor below the temperature at which it functions to drain the collectors. The application of ice to the sensor should turn off the pump and allow the collector to drain. Removal of the ice will then permit the sensor to warm up and restart the pump. These operations can be visually verified.

Proper functioning of controls and other components withch prevent overheating and/or boiling of fluids in the collector and storage loops must also be verified. Several types of overheat protection devices are commonly used, so the manufacturer's instructions should be carefully followed in the check-out procedure. An excessive temperature condition can usually be electrically simulated by a suitable signal to the controller, and the functioning of the overheat protection system observed. Circulation of water from the storage tank through a heat rejection coil, drainage of the collector, draw-off of domestic hot water from the solar pre-heat tank and automatic addition of cold water, and discharge of hot water from the solar hot water storage tank, are the principal methods employed.

Checking the operation of whatever system is in use is essential. If the overheat sensor is accessible, a small electric heating element can be temporarily applied at the proper point by the installer, thereby checking the entire system in place, including the control elements. If this condition has to be simulated by causing an open circuit or short circuit in the overheat sensor contact in the controller, the sensor itself should be checked by heating prior to installation. Controller manufacturers usually provide check data for all standard elements in the control circuitry. Measurement of electrical resistances can usually provide satisfactury evidence of proper operation.

PERFORMANCE CHECK

Following the checking of proper functioning of all components in the system under all conditions of operation, measurement of heat output and efficiency of the system should be made. The following table (Performance Analysis, Liquid Heating System) indicates the measurements needed and the calculations required. The previously described methods for measuring collector flow rate

V-S-425

PERFORMANCE ANALYSIS

Liquid Heating System

Collector:

1. Area, A_c ________ ft^2
2. Inlet temperature, T_i ________ ^oF
3. Outlet temperature, T_o ________ ^oF
4. Pressure drop across pump ________ psi
5. Fluid or flow rate, G ________ gpm
6. Specific heat, c_p ________ $Btu/(lb \cdot {}^oF)$
7. Fluid specific weight, γ ________ lb/gal
8. Heat delivery rate: $Q_c = G\gamma\, c_p\, (T_o - T_i)(60)$ ________ $\frac{Btu}{hr}$

Collector Efficiency:

1. Solar radiation on horizontal surface, I_H, or ________ $Btu/(ft^2 \cdot hr)$
   tilted surface, I_T ________ $Btu/(ft^2 \cdot hr)$
2. Collector efficiency

$$n_c = \frac{Q_c}{I_T A_C} \times 100$$ ________ %

and temperature rise at full and nearly constant solar radiation levels should be conducted simultaneously with measurement of solar radiation intensity. These measurements should be made near mid-day so that conditions are as constant as practical. Calculation of the heat delivery from the collector is a simple multiplication of the temperature rise, flow rate, and heat capacity factors. Dividing this quantity by the solar radiation input rate provides efficiency data. Comparison with the manufacturer's rating data at the temperatures and solar radiation levels corresponding to those applied during the test can then show the degree of agreement with the designed values. Unexplained differences, if any, then need to be investigated and corrected.

INSTALLATION CHECKLIST

Although it is mandatory standard procedure to consult the installation, operations and maintenance instructions supplied by the manufacturer for each spproved system, the following general checklist will serve as a useful guide for installing a typical solar system.

YES	NO	
		1. SITING AND ORIENTATION
_____	_____	1-1 Are collectors oriented in a proper southerly direction?
_____	_____	1-2 Do solar collectors have an unobstructed view in a southerly direction between 9:00 a.m and 3:00 p.m.?
_____	_____	1-3 Are collectors tilted within acceptable limits?
_____	_____	1-4 Are system components located in such a manner as to harmonize with surroundings, to minimize vandalism and obstruction to pedestrian or vehicular traffic, and to facilitate emergency access?
_____	_____	1-5 Are system components located in such a manner as to allow easy access for cleaning, adjusting, servicing, examination, replacement or repair, especially without trespassing on adjoining property?
_____	_____	1-6 Is there safe and easy access to gutters, downspouts, flashing, and caulked joints to allow minor repairs and preventative maintenance.
_____	_____	1-7 Are collectors located to minimize heat losses?
_____	_____	1-8 If ground-mounted, are collectors located to minimize interference from drifting snow, leaves and debris?
_____	_____	1-9 If roof-mounted, are existing roof structures capable of supporting the additional load imposed by the collectors?

2. COLLECTOR MOUNTING

YES	NO	
____	____	2-1 Is the framework constructed to support collectors under anticipated extreme weather conditions (wind loading up to 100 MPH, ice, rain, etc.)?
____	____	2-2 Has the framework been treated to resist corrosion?
____	____	2-3 Are joints between the framework and the rest of the building caulked and/or flashed to prevent water leakage and are collectors installed so as not to contribute to moisture build-up, rotting, or deterioration of the roof or wall of the building?
____	____	2-4 Are collectors installed so that water flowing off warm collector surfaces cannot freeze in cold weather and damage roof or wall surfaces?
____	____	2-5 Have collectors been mounted with weep holes (if provided) at the lowest end of the collector?
____	____	2-6 In areas that have snow loads over 20 pounds per square foot or greater, have provisions been made to deflect snow or ice that may slide off roof-mounted components and endanger vehicles or pedestrians?

3. PIPING AND VALVES

YES	NO	
____	____	3-1 Have the required building, plumbing, and electrical permits (if necessary) been obtained prior to the start of installation?
____	____	3-2 Have solar components been ordered well in advance of the scheduled date to begin instruction?
____	____	3-3 Are connections to potable water lines being made by a licensed plumber where required by local codes?
____	____	3-4 Is all piping properly insulated to maintain system efficiency?

YES	NO		
______	______	3-5	Is all exposed insulation protected from weather damage?
______	______	3-6	Are sufficient pipe hangers, supports, and expansion devices provided to compensate for thermal effects?
______	______	3-7	In ground-mounted systems, are insulated pipes to and from collectors buried below the frost lines?
______	______	3-8	Is piping for draindown systems properly pitched to facilitate draining of fluid from the collectors?
______	______	3-9	If ground-mounted collectors are used, is the run of pipe to storage and back reduced to the absolute minimum?
______	______	3-10	Have isolation valves been provided so that major components of the system (pumps, heat exchangers, storage tank) can be serviced without system draindown?
______	______	3-11	Have air bleed valves been provided at high points in the system so that air can be removed from the liquid circulation loop during both filling and normal operation?
______	______	3-12	Have suitable connections been supplied for filling, flushing, and draining both the collector loop and the potable water piping of the system?
______	______	3-13	Has piping been leak tested to 1-1/2 times system design pressure for at least 1 hour at constant temperature (with collectors covered) prior to backfilling and insulating.
______	______	3-14	Has corrosion between dissimilar metals been avoided by the use of suitable inhibitors in the system as well as dielectric washers in the mounting?

YES	NO		
______	______	3-15	Has care been taken not to short out the insulating effect of dielectric washers between dissimilar metals by pipe hangers, control systems connections, etc.?
______	______	3-16	Will heat transfer fluids be safe and stable at both stagnation temperature and normal running temperatures?
______	______	3-17	If a system using antifreeze is used, have a fill valve and a drain (for sampling) been provided in the collector loop?
______	______	3-18	Has a tempering valve or other temperature limiting device been installed to limit exit temperature of the hot water to a safe level?
______	______	3-19	If a system containing antifreeze is used, have threaded joints been taped with tightly drawn Teflon^R tape?
______	______	3-20	Are all systems, subsystems, and components clearly labeled with appropriate flow direction, fill weight, pressure, temperature and other information useful for servicing or routing maintenance?
______	______	3-21	Are there vacuum relief valves in the system to prevent the collapse of storage or expansion tanks?
______	______	3-22	Has care been taken to install the circulator pumps so that fluid is flowing in the proper direction?
______	______	3-23	Has the expansion tank been located on the suction side of the pump?
______	______	3-24	Has a check valve been installed in the collector loop to prevent reverse circulation by thermosiphoning at night?
______	______	3-25	Are vacuum relief valves protected from freezing?

YES	NO		
		4.	STORAGE TANK
		4-1	Is the storage tank insulated to at least R-11?
		4-2	Are the piping connections to the tank located to promote thermal stratification?
		4-3	If a storage tank is installed on a roof or in an attic, is it provided with a drip pan and an outlet to an adequate drain?
		4-4	Is the storage tank properly connected to the conventional water heater?
		4-5	Are buried storage tanks anchored to prevent flotation in case of high groundwater levels?
		5.	SYSTEM SAFETY
		5-1	Are all surfaces with running temperatures at 120° F, or higher isolated from pedestrian traffic in order to prevent burns?
		5-2	Are temperature and/or pressure relief valves installed so that pedestrians or equipment are not exposed to effects of venting valves?
		5-3	Are temperature and/or pressure relief valves installed so as to prevent system pressures from rising above working pressure and temperatures?
		5-4	When toxic or flammable fluids are used in the system, will fluids overflow or discharge into sewers or storage in a manner acceptable to the local applicable codes?
		5-5	If supplied water pressure is in excess of 80 pounds per square inch or the working pressure rating of any system components, has an approved pressure regulator preceded by an adequate strainer been installed?

YES	NO		
		5-6	Has the system been designed so that any direct connection between wastes from the system and potable water is impossible?
		5-7	Is there an approved backflow preventer at the cold water supply inlet if required?
		5-8	Is there a double-walled heat exchanger in the system or another approved method of separating nonpotable heat transfer fluids from potable water?
		5-9	Have all outlets and faucets on the nonpotable water lines of the system that might be used by mistake for drinking or domestic uses been marked "DANGER - WATER NOT DRINKABLE"?
		5-10	If hazardous fluids are used in the system, have proper procedures for their use, including first-aid, handling and safe disposal been supplied to the owner?
		5-11	Is adequate drainage available in the collector piping array for leaks in collectors and discharges from pressure relief valves?

6. ELECTRICAL SYSTEM

YES	NO		
		6-1	Does field electrical wiring comply with all applicable local codes and equipment manufacturer's recommendations?
		6-2	Is there a properly grounded and protected power outlet for the system controls?
		6-3	Has control circuit wiring been color-coded or otherwise labeled so that wires are readily traceable?
		6-4	Are the sensors for collectors and storage tank attached tightly for the best possible thermal transfer and located per equipment manufacturer's instructions?

YES NO

_____ _____ 6-5 Is the collector temperature sensor located in a collector or near the exit from the collector array?

7. CHECKOUT AND START-UP OF SYSTEM

_____ _____ 7-1 Has a person qualified in both solar and conventional hot water systems put the system through at least one start-up and shutdown cycle, including putting the system through all modes of operation?

_____ _____ 7-2 Has the owner been instructed in the proper start-up and shutdown procedures, including the operation of emergency shutdown devices, and fully instructed in the importance of routine maintenance of the system, including cleaning collector glazing and other components, draining and refilling the system, air venting, corrosion control, and other procedures?

_____ _____ 7-3 Do operating instructions include provisions for the system if the owner leaves for a vacation and hot water use is nil?

_____ _____ 7-4 Has the system been designed so that both solar and conventional systems can operate independently of each other?

GOOD PIPING PRACTICES

There are piping practices that should be applied when plumbing any hydronic system. The following list contains many of those practices. Some of the items may have already been considered when designing the system. The installer should review the list before beginning the piping.

1. Provide adequate shut-off valves and unions to facilitate circuit isolation and component servicing. Also, unions must be located properly for component removal.

2. Provide necessary throttling valves for flow balancing.

   a. Gate valves are not designed for throttling services.

   b. Install the throttling valve downstream of a fixture, whenever possible. (This minimizes turbulence and vapor flushing).

3. Provide adequate instrumentation for determining pressures, temperatures, flows, etc.

   NOTE: Some devices such as circuit-setters may be used for a combination of services such as throttling, flow readout and shut-off.

4. Orient pump volume such that entrapment of air is minimized.

5. Provide adequate supports for pumps and piping to reduce stresses and vibrations.

6. Provide pipe line vibration eliminators and expansion joints as required.

7. Size piping to avoid high velocities which cause noises and require greater pumping power.

8. Allow sufficient spacing around equipment for servicing, such as heat exchanger tube bundle removal.

9. Pitch-up piping in flow direction 1/16" to 1/8" per foot whenever possible.

10. Install non-metallic connectors to isolate electrolytically active dissimilar metal components of the system (such as aluminum collectors with copper tube piping). It is best to minimize usage of dissimilar metals in a system.

11. Avoid bull-head connections and other connections which cause sudden velocity changes.

12. Provide vents at all high points.
    a. Use manual vents if there is a probability of negative (vacuum) pressure at the vent location.
    b. Pipe vent discharges, particularly automatic air vents, to safe locations to prevent water damage.
    c. Locate vents for easy accessibility.

13. Size all downrunner pipes for adequate flow velocity (to entrain air). A minimum velocity of about two feet per second is suggested.

14. Provide an air separation device in a closed-loop piping circuit.

15. Provide an adequately sized compression tank and necessary trim fittings.

16. Provide a system fill connection. Be sure to check local piping codes if the potable water supply is connected directly to the system.

17. Provide necessary drain valves.
    NOTE: In areas conducive to freezing probabilities, provide drain plugs or valves at all low points.

18. Pipe relief valve outlet to safe location. DO NOT thread exit end of drain pipe. Check local piping codes for discharge into floor drain or sewer piping.

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